DESIGN OF A 200 HORSEPOWER AERONAUTICAL ENGINE

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ARMOUR INSTITUTE OF TECHNOLOGY

1920

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DESIGN OF A 200 HORSEPOWER AERONAUTICAL ENGINE

A THESIS

PRESENTED BY

P. E. WINKLESS, G. E. O. PETERSON AND C. F. PETERSON

TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

BACHELOR OF SCIENCE

IN

MECHANICAL ENGINEERING

MAY 27, 1920

APPROVED:

Professor of Mechanical Engineering

Dean of Engineering Studies

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ILLINOIS INSTITUTE OF TECHNOLOGY
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Foreword.

The individual who is not thoroughly conversant with the great improvement that has been made in motor car engines, will have difficulty in appreciating how the great capacity per unit of mass of the light motors designed especially for aerial work is largely possible through experience gained in the automobile engineering field. When aeroplanes were first devised numerous unconventional forms of internal combustion engines were contrived that were believed to have features making them specially suitable for the propulsion of aircraft.

At present there are scores of aeronautical engines, each having its own peculiarities of design; but all following the same general plan. Consequently, in the preparation of this work it was considered logical to follow standard practice rather than deviate from the present designs.

It is hoped that interest in aeronautical engineering will steadily increase to such extent that further developement will result in time to

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come.

The writers desire to acknowledge their special indebtedness to Professor Daniel Roesch for valuable information and suggestions.

P.E.W.
G.E.P.
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OBJECT.

The object of this work was primarily to design the important members of an experimental six cylinder aeronautical engine capable of delivering 200 horsepower at the normal speed of 1750 R.P.M.

The writers, at the outset, realized that the perfection of the aeronautical engine is a matter of exhaustive testing and redesigning and have undertaken the problem with the idea of acquainting themselves with the difficulties encountered by designers of that type of engine.

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Preliminary Considerations.

The problem of the airplane engine should appeal strongly to every engineer because it is a problem of the lightest power plant. The lightest weight of engine proper per horsepower is to be secured first by obtaining maximum mean effective pressure at maximum speed and second by making the weight of metal per cylinder, or per cubic inch of cylinder displacement per working stroke, a minimum. With both these factors in view, the engine must be reliable in operation.

Considering the effect of cylinder diameter upon unit metal weight, it appears that from the unit weight standpoint the cylinder diameter should be as large as possible, because the wall thickness of a cylinder is always greater than necessary for the stress due to structural reasons. This being the case, the larger the cylinder per cubic foot of displacement the less the unit metal weight in the wall, and the only limit to

large cylinder diameter is good operation.

As the stroke is increased, the metal in the cylinder piles up endwise, too fast with reference to volume, and therefore for minimum unit metal weight, the shorter the stroke the better. For that reason as a general thing shorter strokes are used on airplane engines than on automobile engines.

The connecting rod length considerably affects the weight. Clearly, the shorter the connecting rod the shorter the frame, and therefore saving in metal. But, on the other hand by decreasing the length we increase the angularity, which introduces stresses requiring metal thickening in other places.

The number of cylinders should be as large as possible up to the point where the weight of the connecting parts has to be increased. A two cylinder engine has less than twice the weight per cubic foot of displacement than a single-cylinder, for the reason that the number

of end supports for the shaft, etc, is not increased. Similarly a three-cylinder has less than three times, a four-cylinder less than four times, and so on; and the weight per cubic foot of displacement becomes less and less until a certain number of cylinders (about six) is reached where the shaft diameter and the weight of the frame must be increased so as to retain the necessary stiffness, whereupon the saving in weight by multiplication is neutralized.

Another method which helps reduce weight is by overrunning the piston at the lower end of the stroke. The effect on weight is clearly seen in that the overall height of the motor is reduced. Except in special cases where the designer finds it advisable to place a ring at the open end of the piston, the latter may overrun the bore close to 1/6 its length.

We may make use of the piston to reduce weight by another method; namely, by making it convex outwardly. When a high compression

ratio is desired this helps to reduce the compression volume and thereby reduce the height of the cylinder. A convex piston head approaches the spherical form and is therefore strengthened. On the other hand, however, it has greater area exposed to the heat; and since the head is not provided with any cooling means this is a matter of considerable moment.

Another idea which may or may not have anything to do with the reduction of weight is in the method of locking the piston pin to prevent it from scoring the cylinder wall. Some designers favor the use of a single set screw in one boss of the piston while others place a piston ring directly over the center of the piston pin. The latter method has the advantage in that it allows a full floating piston pin which is desired. The Liberty engine uses aluminum caps over the ends of the piston pin. A new idea brought out and at present little used is in placing a steel spring ring in a groove on either

side of the piston pin. This construction is favored in the present design chiefly because of its simplicity and its effect on the reduction of weight.

The weight of an engine is also a function of the vertical drive shaft speed. The magnetos for a six cylinder vertical engine should be run at 1.5 crankshaft speed. Should the vertical shaft be run at crankshaft speed the magnetos must of course be geared up. The housing must therefore be increased in size and an increase at this point means that the whole vertical drive must be moved outward with the resulting increase in weight. By running the vertical shaft at 1.5 engine speed the above difficulty is overcome and in addition, the increase in speed results in the decrease in torque and the shaft may be made lighter.

General Description.

(A) Cylinders.

The cylinders are steel sleeves made from pierced forgings similar to high explosive and shrapnel shells. These are solid at the upper end and the valve seats are formed directly in the bottom of the cylinder itself. The outside of the cylinder is threaded, and screws into an aluminum casting which looks like a cylinder block but is in reality only the water jacket for all six cylinders of the block. The lower ends of the cylinder sleeves project and have flanges fastening the cylinder block to the engine base by means of triangular bridge clamps; the latter being held in place by ½ inch rods which extend through holes in the webs of the two halves of the crankcase. The individual cylinder sleeves are heat treated, machined and threaded on the outside. The cylinders are screwed into the cast aluminum cylinder block

which comprises the water jackets and valve ports, intake and exhaust passages.

(B) Valves and valve gear.

As is before stated, the valves are located in the heads of the cylinder sleeves. They are set vertically in the cylinders along the center of each block and are directly operated by a single camshaft located over the valves. The valves are of chrome-nickel steel, with large diameter hollow stems, working in tight fitting cast iron bushings, provided at the upper ends with case hardened flat headed adjusting discs, upon which the cams operate and are held to their seats by single helical springs. The hollow camshaft is mounted in four plain bronze bearings and is driven from the crankshaft by a vertical shaft and bevel gears of hardened alloy steel. This vertical shaft which runs at 1.5 crankshaft speed is mounted in two combined radial and thrust ball bearings located

at and near the lower end, and in a plain bearing at its top end. Directly above the intermediate ball bearing this shaft carries a bevel pinion which meshes with two other bevel pinions of the same size as itself. The latter pinions, which are mounted on a common axis, drive the two Dixie high-tension magnetos, located on opposite sides of the vertical shaft. The upper part of the vertical shaft is inclosed in a housing of light steel tubing and the lower part in an aluminum housing, the upper end of which is large enough to allow the intermediate radial ball bearing to be put into place. The bevel gears for the magnetos are mounted directly on the shafts of these devises and no extra bearings are required in the housing. Light steel caps are screwed into the housing over the magneto shafts, back of the driving gears and are provided with felt washers to keep out the dust and keep in the oil spray. The hollow camshaft

has an outside diameter of 1 $\frac{3}{8}$ inches and an inside diameter of 1 $\frac{1}{8}$ inches. At the forward end it is provided with an eight tooth spline upon which is mounted the camshaft driving gear. The gear is held in place by a lock nut which is prevented from coming loose by a steel pin. The cams are forged integral with the camshaft, cut and ground to size and hardened. The valves have a lift of $\frac{3}{8}$ in. and have a clear opening of 2 $\frac{1}{8}$ in. The valve discs are adjusted to give a .02 in. clearance when the engine is cold. From the forward end of the camshaft a speedometer drive is taken off.

(C) Pistons.

Die cast aluminum pistons are employed. These are of unusual design in that the piston head and a considerable portion of the skirt are about ½ in. thick, the object in using such heavy walls being, of course, to facilitate the flow of heat from the head to the skirt. Each piston is cut with grooves

for three compression rings near the top end. Depressions are formed on the piston surface where the bosses are located, so that any slight distortion of the bosses will not affect the bearing of the piston in the cylinder. Each piston was estimated to weigh between 3 and 4 lbs. when finished. There are no ribs inside the piston to strengthen the bosses and help to conduct away the heat, these being rendered unnecessary by the heavy wall thickness employed. The bosses are very firmly secured to the skirt, however, by large fillets; in fact, the bosses really form solid segments inside the cylindrical skirts. The piston pin is full floating and is prevented from scoring the cylinder wall by two steel spring rings which fit into grooves in the piston bushings. All of that part of the skirt not occupied by ring grooves has oil grooves cut upon it. The skirt wall is tapered from about the axis of the piston pin to the open end,

while the head is plain. Over the whole length of the skirt the piston is finished up to .02 in. below the nominal cylinder bore, while on the lands, it is finished to diameters which are .025, .030, and .035 in. below the cylinder bore respectively.

(D) Connecting rods.

The connecting rods are tubular and are machined all over from a steel forging. The outside diameter of the tubular rod is $1 \frac{1}{8}$ in. and the inside is bored with a $\frac{27}{32}$ in. hole from the crankpin end to within $\frac{3}{16}$ in. of the wrist pin busing. The bottom end of the bore is fitted with a screwed plug. The weight of the big end is lightened by holes drilled radially through the big end. Two semi-circular oil grooves are machined in the babbitt metal of the big end bearing cap and a circular groove is cut in the wrist pin bushing for lubrication. The bronze wrist pin bushing is forced into the rod and is

broached out to a diameter of $1\frac{1}{4}$ in. The connecting rod cap is held in place by four $\frac{3}{8}$ inch bolts with castellated nuts and split pins. The bore of the big end of the connecting rod is 2 inches and the length of the crankpin bearing is $2\frac{1}{2}$ inches. All the connecting rods are balanced by weighing, the limits being $\pm\frac{1}{4}$ oz.

(E) Crankshaft.

A seven bearing crankshaft is used, having main journals $2\frac{3}{16}$ inches in diameter. All intermediate bearings are $2\frac{1}{16}$ inches long. The forward end bearing is $2\frac{1}{4}$ inches long and the rear end bearing $4\frac{3}{4}$ inches. Three-quarter inch holes are drilled through all of the journals of the crankshaft with the object of lightening it, and also in the case of the crankpins, to help in the lubrication. To a forged flange at the forward end of the crankshaft is bolted the crankshaft gear. The crankshaft is a chrome nickel steel forging and is machined all over. It is

provided with a thrust collar to the rear of the rear bearing, back of which is mounted a double ball thrust bearing. The propeller hub seat is turned to a taper and cut with a keyway. The crankshaft being made of alloy steel is, of course, heat treated.

(F) Crankcase.

The crankcase is in two pieces, both of which are aluminum castings. The crankshaft bearings are on a line with the split in the crankcase, the lower halves of the crankshaft bearings being held in the lower half of the crankcase and the upper halves in the upper half of the crankcase. The two halves are tied together by long bolts which pass through the upper half of the crankcase, through bosses, and clamp the cylinder sleeves in place as before described. This gives an accessible construction which is at the same time rigid. A careful joint is made between the two halves of the crankcase in order to secure the desired alignment

at the main bearings, the joint being lapped.

(G) Lubrication.

The oil supply is carried in a reservoir which is cooled. This reservoir is mounted somewhere in the vicinity of the engine and from it oil is led to the connection on the right side of the oil pump body, which is to be marked "Oil In". The oil is filtered at this point through a large-area, fine mesh screen. A delivery pump of the gear type takes the oil after it has passed through the screen and delivers it under pressure to a distributor pipe running the entire length of the crankcase. There is a pressure regulating valve between the pump and the distributing pipe which should hold the oil pressure so that it does not exceed 50# per sq. inch.

From the distributor pipe there are pipes fitted in the crankcase leading to the main crankshaft bushings. The crankshaft is hollow,

and in the center of each main bearing there is a radial hole drilled through the shaft into the hollow center. A passage leads from each hollow main bearing to the adjacent crankpin, which is also hollow. A radial hole is drilled through each crankpin and carries the oil out on the surface of the pin.

The oil spray thrown off by centrifugal force from the ends of the connecting rods lubricate the piston pins and cylinder walls. A part of the oil from the main distributor pipe at the propeller end of the engine goes up through a pipe that leads to the propeller end camshaft bearing. The oil is lead through a passage drilled diametrically through the bearing midway of its length. Once every revolution of the camshaft, a hole drilled through the camshaft, into its hollow center registers with the oil passage through the bearing. Thus, once every revolution of the camshaft a small quantity of oil is forced into the

hollow center. The oil is led through the camshaft and out through holes drilled in it to each camshaft bearing. A small hole is drilled in the base circle of each cam to properly lubricate the tappet. The excess oil eventually finds its way to the gear end of the camshaft, over the gears and down the driveshaft housing into a chamber just above the oil pump.

The excess oil thrown off in the crankcase by the connecting rods collects in this same chamber when the engine is inclined so that the propeller end is high. If the propeller end of the engine is low, this oil collects in a small sump or chamber at the propeller end of the crankcase.

Immediately above the oil delivery pump is located an oil return pump consisting of three gears and driven by the same shaft as the delivery pump. The function of this oil return pump is to draw the excess oil out of the crankcase and

return it to the oil reservoir. One section of this pump draws oil from the sump at the propeller end of the crankcase and the other section draws oil from the sump at the magneto end of the crankcase. Both halves of the pump deliver oil to the connection on the left side of the oil pump body to be marked "Oil Out" from which point it returns to the oil reservoir.

(H) Cooling system.

The cooling water is circulated by means of a centrifugal pump connected through a spline to the crankshaft gear, and obviously runs at crankshaft speed. The housing of the pump is made of aluminum and the impeller of bronze. As it leaves the pump the water passes through a pipe leading under the magneto base on the left side and connections through a hose joint to the cylinder block.

(I) Propeller Hub.

The new standard hub adopted by the aviation

section of the United States Army is used. The shaft is given a taper of one in ten, and the hub is prevented from turning by a straight key held in place by a flat-headed standard A.S.M.E. machine screw. The hub is forced in place on the taper by the long nut by means of a bar put through the holes at the end of the nut, and the nut is then prevented from turning by the pin, which drops into one of the four slots around the circumference of the nut.

The inner flange is integral with the spool or barrel of the hub, but the outer flange has a movement of 1 inch, allowing for hub variations of from 5 to 6 inches in thickness. The outer flange is prevented from turning by ten splines; and as five of these splines are drilled to receive the locking pin, it gives a large number of positions in which the nut can be securely locked in place. This pin is held in position by a wire running through its head and around the hub, a

groove being provided for this purpose. The ends are twisted together in the usual way.

The bolts have a taper head to insure a perfect fit in the inner flange, and the nut on the outer end has a projecting sleeve that goes through the outer flange, so that the thread of the bolt does not come in contact with the flange at any point. It will be noticed that the engine shaft is hollow and that the bolt is drilled for nearly its entire length. It will also be seen that the key has a tapped hole at the front end through which a screw can be used as a jack in lifting the key out of place, should it become jammed in any way.

Method of Design and Calculations.

(A) Thermodynamic Analysis.

In the design of this engine it was assumed that a bore of 5 inches and a stroke of 7 inches was desired, as is used by the Liberty and Hall Scott airplane engines. The average compression ratio was taken as 5 to 1, and the horsepower rating was to be at the speed of 1750 R.P.M.

Let:

b = bore of cylinder in inches.

L = stroke in inches.

r = compression ratio.

P_1 = initial compression pressure.

P_2 = maximum compression pressure.

V_1 = absolute volume at beginning of compression (piston displacement – clearance volume)

V_2 = absolute volume at end of compression (clearance volume)

n = exponent for adiabatic change of state.

From the law of adiabatic change we have:

$$P_1 \; V_1^n = P_2 \; V_2^n = \text{Const.}$$

From which it follows that the compression pressure

$$P_2 = P_1 \left(\frac{V_1}{V_2}\right)^n$$

But $\frac{V_1}{V_2} = r.$

Therefore:

$$P_2 = P_1 \, (r)^n$$

The initial pressure P_1 varies with the piston speed, size and form of passages, and with other factors. At low engine speeds P_1 is generally close to atmospheric pressure but at rated engine speed it drops off so that it is not more than from 12 to 13 pounds per square inch. For a good filling engine we may safely assume that P_1 is 1.5 lbs below atmospheric (i.e $P_1 = 14.7 - 1.5 =$ 13.2 lbs per sq. in.) The value of n for compression is an experimental quantity which is found from the actual indicator diagram and may be taken as 1.30. Substituting known values we obtain for

the final compression pressure $P_2 = 13.2 \times (5)^{1.30} =$ 106.9 lbs per sq. in. The explosion pressure varies with different shapes of combustion chambers and in this case the ratio may be taken as 4.25 to 1. We obtain, therefore, as the maximum explosion pressure,

$$106.9 \times 4.25 = 454. \text{ lbs per sq. in.}$$

The work of expansion as given by Heldt is:

$$W_e = \frac{\pi b^2 a P_2}{14.4} \left(\frac{L}{r - 1}\right) \left[1 - \left(\frac{1}{r}\right)^{.32}\right]$$

Where:

b = bore of cylinder in inches.

a = ratio of maximum pressure to compression pressure.

P_2 = maximum compression pressure given in lbs per sq. in.

L = length of stroke in inches.

r = compression ratio.

Substituting the numerical values we obtain:

$$W_e = \frac{\pi \times (5)^2 \times 4.25 \times 106.9}{14.4} \left(\frac{7}{5 - 1}\right) \left[1 - \left(\frac{1}{5}\right)^{.32}\right]$$

$$= 1744. \text{ foot lbs.}$$

The work of compression as given by Heldt is:

$$W_c = \frac{\pi b^2 P_1}{14.4} \left(\frac{rL}{r-1}\right)\left(r^{0.3} - 1\right)$$

Where P_1 = initial compression pressure lb. per sq. in.

Other notation same as before.

Substituting the numerical values we obtain:

$$W_c = \frac{\pi \times (5)^2 \times 13.2}{14.4} \left(\frac{5 \times 7}{5-1}\right)\left(5^{0.3} - 1\right)$$

$= 391.$ foot lbs.

The useful work per cycle is therefore:

$W_e - W_c = (1744 - 391.) = 1353.$ foot lbs.

At 1750 R.P.M. there are 875 explosions per minute and the work done by each cylinder per minute is,

875 X 1353 = 1,183,875 foot lbs.

One horsepower 33,000 ft. lbs per min. so that each cylinder develops, $\frac{1,183,875}{33000} = 35.9$ Horsepower.

Since the engine has six cylinders, it develops,

35.9 X 6 = 215.4 Horsepower.

The total M.E.P. on each piston is,

$$\frac{\text{net output of the cylinder}}{\text{stroke of engine in feet}} = \frac{1353}{\left(\frac{7}{12}\right)}$$

= 2320 lbs.

The area of the piston is,

$$\frac{\pi}{4} d^2 = \frac{\pi}{4} (5)^2 = 19.635 \text{ sq. inches.}$$

and the M.E.P. is, therefore,

$$\frac{2320}{19.635} = 118.3 \text{ lbs. per sq. in.}$$

The above analysis is purely theoretical and does not account for the inherent losses prominent in any engine. The values, therefore, in the actual engine are somewhat reduced.

(B) Cylinder Design.

(1.) Clearance volume.

Let:

b = bore of cylinder in inches.

L = length of stroke in inches.

r = compression ratio.

V = piston displacement cu. inches.

C = clearance volume cu. inches.

Then,

$$r = \frac{V + C}{C}$$

or

$$C = \frac{V}{r - 1}$$

But

$$V = \frac{\pi}{4} b^2 L$$

and therefore,

$$C = \frac{\frac{\pi}{4} b^2 L}{r - 1} = \frac{\pi b^2 L}{4(r - 1)}$$

Substituting the numerical values,

$$C = \frac{\pi X (5)^2 X 7}{4(5 - 1)} = 34.4 \text{ cu. inches.}$$

The shape of the clearance space and the piston head are shown in Fig. 1. The problem now is to find the height h, , given the piston head dimensions and the clearance volume.

It can readily be seen from similar triangles that h = 1 inch.

The area of a 5 inch diameter circle is 19.635 sq. in. The area of a $2\frac{1}{2}$ inch diameter circle is 4.908 sq. in. The volume of the cone of height h is,

$$\frac{1}{3} X 19.635 X 1 = 6.545 \text{ cu. inches.}$$

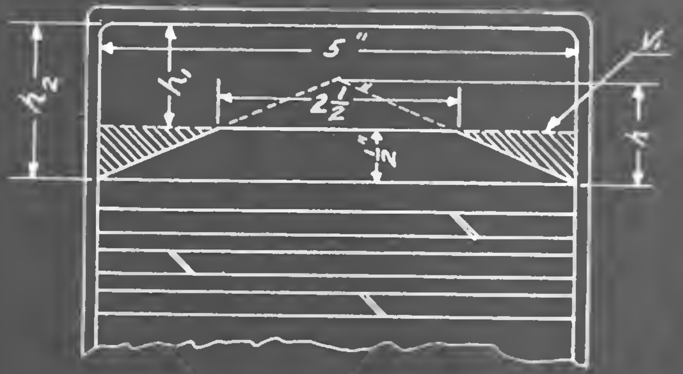

5"
h_2
h_1
$2\frac{1}{2}$
V_1
h
$\frac{1}{2}$"

FIG. 1.

The volume of the small cone is,

$$\frac{1}{3} \times 4.908 \times \frac{1}{2} = .818 \text{ cu. inches.}$$

Therefore the volume of the frustum of the piston head is, (6.545 − .818) = 5.727 cu. inches.

The volume of the cylinder of the same height as the frustum is, $19.635 \times \frac{1}{2} = 9.8175$ cu. inches.

Therefore the volume represented by the shaded area of Fig. 1, is, (9.8175 − 5.727) = 4.0905 cu. inches and the difference between the clearance volume and the volume represented by the shaded area must equal the volume of height h_1.

Therefore,

$$h_1 = \frac{(34.4 - 4.09)}{19.635} = 1.543 \text{ inches.}$$

The nearest fraction to this is $1 \frac{35}{64}$ inches.

The height h_2 is, therefore,

$$1 \frac{35}{64} + \frac{1}{2} = 2 \frac{3}{64} \text{ inches.}$$

(2.) Thickness of cylinder liner.

The cylinder liner is made of heat treated steel. A conservative ultimate tensile stress of 60,000 lbs per sq. inch is used for the material. In the

Maybach Aviation engine the safe stress used is 13,280 lbs per sq. in. for the longitudinal seam; and, based on the assumed ultimate stress of 60,000 lbs, the factor of safety is,

$$F = \frac{S_u}{S_s} = \frac{60000}{13280} = 4.52$$

We may assume that the maximum unit pressure in the cylinder is 450 lbs per sq. inch for a compression ratio of 5 to 1.

The equation for the strength of the cylinder in longitudinal tension is,

$$t = \frac{PD}{2S_t}$$

Where:

t = thickness of cylinder in inches.

P = unit pressure lb. per sq. in.

D = diameter of cylinder in inches.

S_t = safe tensile stress lbs. per sq. in.

Substituting the values,

$$t = \frac{450 \text{ X } 5}{2 \text{ X } 13,280} = .0847 \text{ inches.}$$

The nearest fraction to this is $\frac{3}{32}$ inch.

The steel liner screws into the aluminum cylinder

block which is $\frac{1}{4}$ inch thick. It is obvious that the aluminum takes some of the stress so that the actual unit stress in the liner is thereby reduced.

Using the practical rule given by Heldt, the width of the water space for engines having bores between 4 and 5 inches, is $\frac{5}{8}$ inch. Foundry practice limits the jacket wall thickness, the smallest practical thickness for aluminum being $\frac{3}{16}$ inch.

(C) Design of Cylinder Bolts.

Four bolts hold the explosion load of each cylinder. The bolts are in direct tension. The total cylinder explosion load at a unit pressure of 450 lbs per sq. in. is,

$$P_1 = \frac{\pi}{4} \times (5)^2 \times 450 = 8,836 \text{ lbs.}$$

The load on each bolt is therefore,

$$P = \frac{P_1}{4} = \frac{8836}{4} = 2,209 \text{ lbs.}$$

A safe stress of 15,000 lbs per sq. inch may be used since the bolts are made from .35 carbon-

chrome-nickel steel, heat treated.

Therefore,

$$S_t = \frac{P}{A}$$

From which,

$$A = \frac{P}{S_t} = \frac{2209}{15000} = .1473 \text{ sq. inches.}$$

This is the area required at the bottom of the thread. From the table of S.A.E. standard screw threads the nominal diameter of each bolt is $\frac{1}{2}$ inch.

(D) Design of Piston and Piston Rings.

The Liberty type piston was used in this design and was made 5 inches in length. The width of the piston rings was made,

$$W = \frac{b}{20} = \frac{5}{20} = \frac{1}{4} \text{ inch.}$$

The thickness opposite the cut was made,

$$t = \frac{b}{24} = \frac{5}{24} = .2085", \text{ say } \frac{3}{16} \text{ inch.}$$

The best location of piston pin. As derived by Heldt, the equation for the best location of the piston pin in the piston is given by,

$$X = \frac{5\ L^2 + .05\ b^2 - .25\ bL.}{10\ L - b}$$

Where,

L = length of piston in inches.

X = distance from piston pin axis to head end of piston in inches.

b = bore of cylinder in inches.

Substituting numerical values,

$$X = \frac{5 \text{ X } (5)^2 + .05 \text{ X } (5)^2 - .25 \text{ X } 5 \text{ X } 5}{(10 \text{ X } 5) - 5}$$

$$= 2.67 \text{ inches, say } 2\,\frac{5}{8} \text{ inches.}$$

The crown of the piston and a considerable portion of the skirt is made $\frac{1}{2}$ inch thick, the object being to facilitate the flow of heat from the head to the skirt.

(E) Design of Piston Pin.

The assumed outside diameter taken from the Liberty motor is 1 $\frac{1}{4}$ inches.

The total load on the piston as previously determined is 8836 lbs at the maximum unit explosion pressure of 450 lbs. Allowing side clearance of the connecting rod between the piston bosses we obtain the length of the connecting rod bearing at

the wrist pin which is 2 $\frac{3}{8}$ inches. The projected area of the bearing is therefore,

$$(1.25 \times 2.375) = 2.97 \text{ sq. inches.}$$

The unit bearing pressure is therefore,

$$\frac{8836}{2.97} = 2{,}980 \text{ lbs. per sq. in.}$$

Heldt recommends 2500 lbs per sq. inch and the Maybach Aviation engine uses 2800 lbs per sq. in. at 450 lbs explosion pressure. The value 2980 lbs per sq. inch exceeds the values given but since the wrist pin is of the full floating type the bearing pressure is reduced considerably by the extra bearing area of the piston bushings, so that the value may be considered a conservative one.

The equation for the determination of the inside diameter of the piston pin is derived by Heldt and is,

$$d_i = \sqrt[4]{d^4 - \frac{d\ b^3\ p}{2s}}$$

Where,

d_i = inside diameter of pin in inches.

d = outside diameter of pin in inches.

b = bore of cylinder in inches.

p = unit explosion pressure lbs per sq. in.

s = safe working stress lbs per sq. in.

The explosion pressure p taken throughout this design is 450 lbs per sq. inch and the value of s may be 20,000 lbs per sq. inch for this high grade steel.

Substituting the numerical values,

$$d_i = \sqrt[4]{(1.25)^4 - \frac{1.25 \times (5)^3 \times 450}{2 \times 20{,}000}} = .910 \text{ inches,}$$

call it $\frac{15}{16}$ inch.

(F) Design of Crankshaft.

The type used follows standard aviation engine practice for six and twelve cylinder engines; namely, the use of seven bearings and hollow bearings and crankpins.

1. Crankpin diameter and length.

An analysis of a number of six cylinder aviation motors (Benz, Hall Scott, Mercedes, Maybach, etc) gave an emperical formula for determining the outside

diameter of the crankpin as,

$$d = \sqrt[2]{\frac{D}{34.6}}$$

Where,

d = outside diameter of crankpin in inches.

D = piston displacement in cubic inches.

For this engine, the piston displacement is,

$$\frac{\pi}{4}(5)^2 \times 7 = 137.4 \text{ cubic inches.}$$

The outside diameter of the crankpin is therefore,

$$d = \sqrt[2]{\frac{137.4}{34.6}} = 1.998 \text{ inches.}$$

call it 2 inches.

The explosion load on the crankpin as predetermined is 8,836 lbs. The bearing pressure for crankpins as given by various authorities is 1800 lbs per sq. inch.

Using this value we find as the length of the crankpin,

$$L = \frac{p}{1800d} = \frac{8836}{1800 \times 2} = 2.455 \text{ inches.}$$

call it 2½ inches.

2. Main journals.

We may assume the thickness of the crank

webs as $\frac{7}{8}$ inch. The center distance between adjacent cylinders is found to be $6\frac{5}{16}$ inches. The length of each intermediate main journal is therefore,

$$6\frac{5}{16} - (2 \times \frac{7}{8}) - 2\frac{1}{2} = 2\frac{1}{16} \text{ inches.}$$

On examination of a number of "six" cylinder aviation engines it was found that the following equations hold true,

$$(1)\quad T = \frac{63025\ H}{N}$$

$$(2)\quad S_s = \frac{2Ta}{\pi(a^4 - b^4)}$$

Where,

H = brake horsepower of engine

N = R.P.M. of engine.

T = torque in inch. lbs.

S_s = shearing stress next to propeller hub.

a = outside radius of journal inches.

b = inside radius of journal in inches.

The average value of S_s in the engines examined was 4,000 lbs. per sq. inch. It was found the better method was to design the journals first

for bearing pressure and then solve by the above method for the inside diameter of the hollow journal.

The average value of the journal bearing pressure was taken as 1000 lbs per sq. inch and the explosion pressure 450 lbs per sq. inch. Two journals carry the load on each piston which is 8,836 lbs. Each journal carries,

$$\frac{8836}{2} = 4418 \text{ lbs.}$$

Since the length was found to be $2 \frac{1}{16}$ inches it follows that the diameter based on bearing pressure must be,

$$\frac{4418}{1000 \times 2.0625} = 2.14 \text{ inches.}$$

call it $2 \frac{3}{16}$ inches.

At the normal horsepower rating of 200 the torque assumed as constant from equation (1.) is,

$$T = \frac{63,025 \times 200}{1,750} = 7,200. \text{ inch lbs.}$$

Substituting the known values in equation (2.),

$$\frac{2 \times 7200 \times 1.094}{\pi \left[(1.094)^4 - b^4\right]} = 4000$$

and solving,

$b = .651$ inches.

The inside diameter is,

$2b = 2 \times .651 = 1.302$ inches.

In the actual design, following Hall Scott six cylinder practice, the inside diameter of all journals and crankpins was made $\frac{3}{4}$ inch.

(3.) Crank cheeks.

The thickness of the cheeks as before assumed was $\frac{7}{8}$ inch. For the width, the following equation was solved for S and this value computed from known dimensions in a number of six cylinder aviation engines.

$$M = S \frac{I}{c} = S \frac{bh^2}{6}$$

And $M = \frac{P}{2} R$ since each cheek carries one half the load.

P = total load on piston in lbs.

R = radius of crank in inches.

b = thickness of cheek in inches.

h = width of cheek in inches.

The average value of S was found to be 12,500 lbs per sq. inch.

Substituting known values,

$$\frac{8836}{2} \times 3.5 = 12,500 \times \frac{.875\ h^2}{6}$$

And h = 2.91 inches.

Call it $2\frac{15}{16}$ inches.

(G) Design of Connecting Rod.

(1.) Design of column.

The connecting rods used are tubular and follow closely the Benz design. The length was taken as 12 inches and the inside diameter of column was taken as $\frac{3}{4}$ of the outside diameter. Using the method given by Heldt, based on the Rankine formula for the strength of columns, we have

$$\frac{P}{0.344\ D^2} + .000526\ \frac{P}{0.344\ D^2} \times \frac{L^2}{.0974\ D^2} = S_c$$

Where,

P = total explosion load on piston in lbs.

D = outside diameter of rod in inches.

L = length between centers of rod in inches.

S_c = safe working stress lb. per sq. inch.

The value of S calculated from the Benz rod was 33,800 lbs per sq. inch, so that,

$$\frac{8836}{0.344\ D^2} + .000526 \times \frac{8836}{0.344\ D^2} \times \frac{(12)^2}{.0974\ D^2} = 33,800$$

solving, we get

$D = 1.112$ inches

Call it $1\frac{1}{8}$ inches.

The inside diameter is therefore,

$\frac{3}{4} \times 1.125 = .844$ inch.

Call it $\frac{27}{32}$ inch.

(2.) Connecting rod bolts.

Four bolts per rod were assumed. The greatest stress comes on the bolts when the piston starts on the suction stroke while the motor runs at racing speed, say, 2000 R.P.M. The force of acceleration is therefore, (See Heldt P-44, Vol. 1).

$$F_a = .0000142\ WLN^2 (\cos \theta + \frac{1}{2n} \cos 2\theta) \text{ lbs.}$$

Where,

W = weight of reciprocating parts in lbs.

$n = \frac{\text{connecting rod length (inches)}}{\text{length of stroke (inches)}}$

L = length of stroke in inches.

N = R.P.M.

It is seen that F_a is maximum when $\theta = 0$

Therefore,

$\cos\theta = \cos 2\theta = I$ when $\theta = 0$ so that the expression $(\cos\theta + \frac{I}{2n}\cos 2\theta)$ becomes,

$(I + \frac{I}{2\left(\frac{12}{7}\right)} \times I) = 1.292.$

The reciprocating weight W includes the weight of the piston and rings, the piston pin and one half the weight of the connecting rod. W for this engine was estimated as $8\frac{1}{2}$ lbs.

Substituting known values,

$$F_a = .0000142 \times 8.5 \times 7 \times (2000)^2 \times 1.292$$

$$= 4370. \text{ lbs.}$$

Part of the rod is also rotating and this part is subject to centrifugal force. The centrifugal force must be added to the accelerating force.

The rotating weight of the rod was estimated to be 5.625 lbs. The radius of rotation of this weight at the moment when the crank is at the top dead center position was estimated to be 3 inches.

Using the equation for centrifugal force for any revolving mass,

$$F_c = 1.226\ WN^2 r$$

Where,

W = rotating weight in lbs.

N = revolutions per second.

r = radius of rotation in feet.

We have therefore,

$$F_c = 1.226 \times 5.625 \times \left(\frac{2000}{60}\right)^2 \times \frac{3}{12}$$

$$= 1915.\ \text{lbs.}$$

The total load on the cap bolts is evidently,

$$F_t = F_a + F_c = (4370 + 1915) = 6,285\ \text{lbs.}$$

A safe stress of 20,000 lbs per sq. inch may be used so that the area at the bottom of the thread of each bolt is,

$$A = \frac{F_t}{4S} = \frac{6285}{4 \times 20,000} = .0785\ \text{sq. inches.}$$

From the bolt table it is found that the nearest bolt corresponding to the above area has a nominal diameter of $\frac{3}{8}$ inch.

(H.) Valve and Valve Gear Design.

(1.) Miscellaneous data concerning valve gear design.

The three possible firing orders for six cylinder engines are:

1. 1 - 4 - 2 - 6 - 3 - 5
2. 1 - 5 - 3 - 6 - 2 - 4
3. 1 - 3 - 2 - 6 - 4 - 5

The firing order chosen is the one used by the majority of six cylinder aviation engines and is,

1 - 5 - 3 - 6 - 2 - 4

Cylinder number 1, is nearest the propeller end of the engine. A six cylinder 4 cycle motor must fire three times per revolution or every 120 degrees of crankshaft travel. Since the camshaft travels $\frac{1}{2}$ crankshaft speed the motor must fire every 60 degrees of camshaft travel.

The direction of rotation of the engine, facing the propeller is anti-clockwise. Due to the valve gear arrangement, the camshaft must travel in the

same direction.

The valve timing was taken from the Liberty motor and is as follows:

Inlet opens, degrees on crank, 10° Late.
Inlet closes, " " ", 45° Late.
Exhaust opens, " " ", 50° Early,
Exhaust closes, " " ", 10° Late.
Ignition, magnetos fully advance, 30° Early.
Ignition, magnetos fully retarded, 10° Late.

The inlet valve is therefore open for a period of 215 degrees of crankshaft travel, or 107 $\frac{1}{2}$ degrees of camshaft travel.

The exhaust valve is open for a period of 240 degrees of crankshaft travel or 120 degrees of camshaft travel. The compression period with spark advanced is 105 degrees crankshaft travel and with spark retarded is 145 degrees crankshaft travel. The expansion period with spark advanced is 160 degrees of crankshaft travel and with spark retarded is 120 degrees of crankshaft travel.

(2.) Valve design.

From the law of continuity of flow,

$$A_P V_P = A_V V_G$$

Where,

A_P = area of piston in sq. inches.

A_V = area of clear opening of valve in inches.

V_P = maximum piston velocity ft. per minute.

V_g = velocity of gas through valve ft. per minute.

Now $A_P = \frac{\pi}{4} b^2$

and $A_V = \frac{\pi}{4} d^2$

where b and d are the diameters of the cylinder and valve respectively.

Therefore,

$$\frac{\pi}{4} b^2 \; V_P = \frac{\pi}{4} d^2 \; V_g$$

And,

$$d = b\sqrt[2]{\frac{V_P}{V_g}}$$

The maximum piston velocity is given by,

$$V_P = 2 \; L \; n.$$

Where,

L = length of stroke in feet.

n = revolutions per minute.

Therefore,

$$V_p = 2 \times \frac{7}{12} \times 1750 = 2040 \text{ ft. per minute.}$$

The value of V_g ranges from 7000 to 9000 ft. per minute. Using 9000 for V_g we obtain,

$$d = 5\sqrt[2]{\frac{2040}{9000}} = 2.37 \text{ inches.}$$

From the layout it was found that the largest valve that can be placed in this type of cylinder was $2\frac{1}{8}$ inches clear diameter. The general dimensions of the valve was taken from practice for engines of this size.

The equation for valve lift was taken from Heldt and agrees closely with practice,

$$h = \frac{d}{8} + \frac{1}{8} \text{ inch.}$$

Where h = lift in inches.

d = clear diameter of valve in inches.

Therefore:

$$h = \frac{2.125}{8} + \frac{1}{8} = .39 \text{ inch.}$$

Call it $\frac{3}{8}$ inch.

(3.) Camshaft design.

The type used is of hollow construction having mushroom cams acting directly on the valve stem discs.

The outside diameter was taken as 1 $\frac{3}{8}$ inch.

Let:

D = center distance between base and top circles of intake and exhaust cams, in inches.

R = radius of base circle in inches.

C = clearance in inches.

r = radius of top circle in inches.

L = lift of valve in inches.

R was taken as $\frac{3}{4}$ inch and r assumed to be $\frac{1}{16}$ inch. The clearance used on both cams was .02 inch.

Then,

$$D = R + C + L - r$$

or

$$D = .75 + .02 + .375 - .0625$$

$$= 1.0825 \text{ inches.}$$

The method of laying out this type of cam is described as follows: (See Fig. 2)

Draw the base circle and top circle, the radii of which have been assumed. The two centers are placed at a distance D apart, as given above. Lay off the clearance circle C outside the base circle. Lay off from the center line connecting the two centers, an angle equal to $\frac{1}{2}$ the valve period. Let this line intersect the clearance circle at the point P. From P draw a tangent T to the clearance circle. Now draw an arc A which is tangent to the base circle, the top circle and the tangent T to the clearance circle. If the arc A becomes straight the radii R and r must be changed.

Drawing #17 shows the valve lift diagrams for both the inlet and exhaust cams. The curves were plotted directly from the cam layout given on drawing number 9.

The advantage in using the mushroom type cam is that the valve accelerates very quickly at the

Page #43.

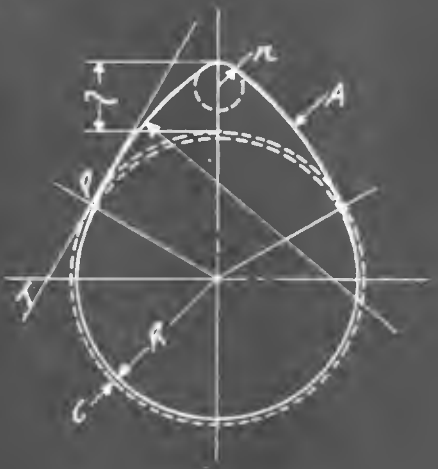


FIG. 2.

beginning of the lift. As compared with the tangential and constant acceleration cams it ranks first in quick opening; but the steepness at the beginning of the lift curve is a measure of the blow dealt the cam follower by the cam when first coming in contact with it. On comparing the lift diagrams of the mushroom, tangential and constant acceleration cams it can be seen that the mushroom type cam crosses the clearance line by far the greatest angle, consequently this cam would be expected to be the most noisy. However, in the mushroom cam the speed of lift at the beginning exceeds that of the other two cams, hence the volumetric efficiency would be expected to be greatest with that type of cam. From a standpoint of strength of valve spring required the mushroom cam ranks first in merit.

From drawing #17, it can be seen that the valve accelerates to the point A and from there to full lift it is retarded. It is also evident that the

lift curve is a time-space curve. Thus, we may use the abscissae as time in seconds, since the crankshaft travel in degrees is a function of the time; and the ordinates as feet, by dividing the lift in inches by 12.

A tangent to the time-space curve will give the instantaneous velocity in feet per second at that particular point, since by definition, $v = \tan\alpha = \frac{ds}{dt}$ also, by definition, the instantaneous acceleration is given by, $a = \tan\beta = \frac{dv}{dt} = \frac{d^2s}{dt^2}$ and therefore a tangent to the velocity curve would give the instantaneous acceleration of the valve, in feet per second, per second. Note: Acceleration always given in feet per second, per second.

(3.-A) Dimensions of camshaft.

The camshaft must be stiff, well supported and have very little deflection under the valve load. When the cam is just at the point of lifting the valve, the latter receives a shock due to

impact and this is, of course, greatest in the case of the exhaust valve. The exhaust valve must be lifted against the force due to inertia of the valve, combined with the initial spring pressure and the pressure of the gases in the cylinder near the end of the expansion stroke. The latter pressure according to Heldt is 50 lbs. per sq. inch.

In the case of the inlet valve only the inertia of the valve and the spring tension need be considered, as the valve does not have to be lifted against a pressure. According to Heldt the force required to lift the exhaust valve is approximately 75 lbs. per sq. inch of valve area.

Referring to Fig. 3,

Let

L = distance between adjacent bearings.

xL = distance of the centre of the cam from the farthest bearing.

P = load on cam in lbs.

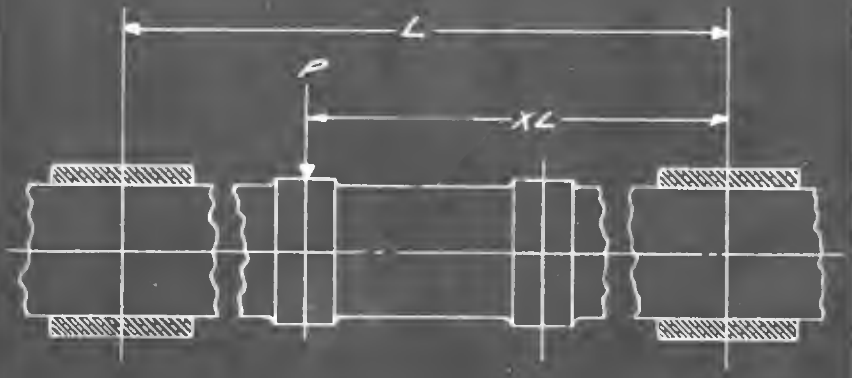


FIG. 3.

E = coefficient of elasticity.

I = moment of inertia of section of shaft.

Then, the deflection of the shaft at the centre of the cam under the total load P will be

$$y = \frac{PL^3}{6EI}\,(2x^4 + 2x^2 - 4x^3)$$

$$\text{or } I = \frac{PL^3}{6Ey}\,(2x^4 + 2x^2 - 4x^3)$$

From the layout we have,

L = 12.625 inches.

xL = 10.6875 inches.

And therefore, x = .8475.

From the table given by Heldt (Page 279) we find for the value of x = .85 that $(2x^4 + 2x^2 - 4x^3)$ = .033.

The deflection, y, should not exceed .002 inch.

$$P = \frac{\pi}{4}(2\tfrac{1}{8})^2 \times 75 = 266 \text{ lbs for the exhaust valve.}$$

Therefore,

$$I = \frac{266 \times (12.625)^3 \times .033}{6 \times 30{,}000{,}000 \times .002}$$

$$= .0490$$

This is the required moment of inertia determined

by the exhaust valve.

For the inlet valve, assume the maximum P equal to twice the mean spring pressure or,

$P = 2 \times 70 = 140$ lbs.

$L = 12.625$

$xL = 8.1875$

and $x = .649$

From the table, $(2x^4 + 2x^2 - 4x^3) = .104$.

Therefore,

$$I = \frac{140 \times (12.625)^3 \times .104}{6 \times 30{,}000{,}000 \times .002} = .0815$$

This is the required moment of inertia determined by the inlet valve. It is seen that the inlet cam determines the size of the camshaft rather than the exhaust cam, in this case.

The moment of inertia of a hollow shaft is given by,

$$I = \frac{\pi}{64}\left[D^4 - d^4\right] = .0491\left[D^4 - d^4\right]$$

Where,

D = outside diameter of shaft in inches.

d = inside diameter of shaft in inches.

Since we assumed that D was $1\frac{3}{8}$ inch and found the

value of I to be .0815 we find that,

$$d = \sqrt[4]{D^4 - \frac{I}{.0491}} = \sqrt[4]{(1.375)^4 - \left(\frac{.0815}{.0491}\right)}$$

$$= 1.18 \text{ inches.}$$

The largest practical dimension that can be used is 1 $\frac{1}{8}$ inches. As a comparison it might be stated that the Hispano Suiza Aviation engine has a camshaft, 34 m/m outside diameter and 28 m/m inside diameter.

The shell thickness is therefore 3 m/m or .118 inch as compared to $\frac{1}{8}$ inch for this design.

(4.) Valve spring design.

The equations used in the calculation of valve springs are:

$$1.\ W = \pi \frac{Sd^3}{8D}$$

$$2.\ F = \frac{8n\ PD^3}{Ed^4}$$

Where:

D = mean diameter of coil in inches.

W = maximum safe load in lbs.

F = compression of spring in inches.

d = diameter of wire in inches.

n = number of coils in spring.

S = maximum safe fibre stress of material lbs per sq. inch.

E = torsional modulus of elasticity.

P = load in lbs.

We may assume E as 12,000,000 and S as 50,000 lbs. per sq. inch. The lift of the valve is $\frac{3}{8}$ inch and the mean diameter of the spring was taken as 1 $\frac{23}{32}$ inches. The spring is to be designed so that it exerts 60 lbs. pressure when the valve is closed and 76 lbs when fully lifted. Since the lift of the valve is $\frac{3}{8}$ inch, 16 lbs pressure compresses the spring $\frac{3}{8}$ inch.

The necessary diameter of wire is found from equation (1.),

$$76 = \frac{\pi \times 50{,}000 \times d^3}{8 \times 1.719}$$

And d = 0.188 inch.

This is approximately the diameter of a No. 7 (Birmingham gauge) steel wire. The actual diameter is .180 inch.

Using equation (2.) we find,

$$\frac{3}{8} = \frac{8 \times n \times 16 \times (1.719)^3}{12,000,000 \times (.180)^4}$$

and

$$n = 7 \text{ coils (approx)}$$

The design was somewhat changed in that d was made No. 9 GA. and n was made 8. When the spring is compressed there is sufficient space between coils so that they do not come together.

(5.) Gear design.

The pitch diameters of gears chosen are as follows,

Camshaft gear - $5\frac{1}{4}$ inches.

Upper vertical shaft gear - $1\frac{3}{4}$ inches.

Lower vertical shaft gear - 3 inches.

Crankshaft gear - $4\frac{1}{2}$ inches.

Magneto gears - $1\frac{3}{4}$ inches.

By taking moments and using the maximum load on cam as before determined we find that the camshaft gear tooth load is 116 lbs. Using equation 1, Page 112 of Nachman's Machine Design,

$$P = \frac{16.8\ W}{S\ b}$$

Where

W = load on tooth in lbs.

S = safe working stress.

b = face of gear in inches.

P = circular pitch of gear in inches.

b for all the drive gears except magneto gears was made $\frac{1}{2}$ inch and S may be taken as 15,000 lbs per sq. inch for this alloy steel.

Therefore,

$$P = \frac{16.8 \times 116}{15000 \times .5} = .260 \text{ inch.}$$

The pitch circumference of the camshaft gear is 16.4934 inches. Therefore the number of teeth is, $\frac{16.4934}{.260} = 63.5$

say 63 teeth.

The upper vertical shaft gear therefore has 21 teeth. This corresponds to a diametral pitch of, $\frac{63}{5.25} = 12$. 12 - 14 pitch stub tooth gears were used. The lower gear set was calculated in identically the same manner and the results are,

Lower vertical shaft gear - 30 teeth.

Crankshaft gear - 45 teeth.

Diametral pitch - 10

10 - 12 pitch stub tooth gears were used. The reason for making the lower gears larger in diametral pitch than the upper set is due to the fact that they operate the oil pump, in addition to the vertical shaft.

(6.) Design of vertical shaft.

This shaft is of hollow construction with the upper vertical drive gear integral.

The maximum torque transmitted by the shaft is small since it runs at 1.5 crankshaft speed.

The torque is:

$$T = PR$$

Where:

P = tooth pressure on upper gear.

R = pitch radius of gear in inches.

Then, $T = 116 \times \frac{1.75}{2} = 101.5$ inch lbs.

For a hollow circular shaft of outside radius a,

and inside radius b,

$$S_s = \frac{2\ Ta}{\pi(a^4 - b^4)}$$

and,

$$b = \sqrt[4]{a^4 - \frac{2\ Ta}{\pi S_s}}$$

The outside diameter was assumed as $\frac{7}{8}$ inch so that $a = \frac{.875}{2} = .4375$ inch.

The safe working stress may be taken as 10,000 lbs per sq. inch.

Hence,

$$b = \sqrt[4]{(.4375)^4 - \frac{2 \times 101.5 \times .4375}{\pi \times 10000}}$$

$$= .428 \text{ inch.}$$

The inside diameter is therefore .856 inch. It is obvious that this shaft is under a feeble stress and one of practical dimensions must be chosen. The inside diameter was made $\frac{11}{16}$ inch.

(7.) Valve timing adjustment.

After assembly it is necessary to time the engine due to side play in the bevel gear train. The usual method is by means of a timing disc placed over the crankshaft at the propeller end.

Place the engine at the upper dead center of cylinder number 1, nearest the propeller end and mount the timing disc on the taper of the crankshaft. Turn the engine in the running direction 10 degrees, (Position where the inlet opens and exhaust closes of cylinder number 1) and put the camshaft in place, the closing of the exhaust cam and opening of the inlet valve cam making the same angle in relation to the valve adjustment mushrooms of cylinder number 1. Tighten the nuts on the four bearings and adjust the proper valve clearance. Turn the crankshaft slowly backward, then gently rotate forward, feeling the exhaust valve which should close 10 degrees after dead center, while at the same time the inlet valve should begin to unseat.

If the timing is not accurate, it is necessary to take the camshaft gear off for the sake of shifting splines, or in turning the vertical shaft, to give the following:

There are 63 teeth in gear and 21 in pinion, 8 spline keys in gear and 5 in pinion shaft. 1 tooth of camshaft gear = 2.857 degrees of camshaft travel or 5.714 degrees of crankshaft travel.

1. Rotate gear clockwise 1 spline key and pinion clockwise $\frac{1}{4}$ tooth to give $\frac{1}{4}$ tooth on camshaft gear advance.

2. Rotate gear clockwise 2 spline keys and pinion clockwise $\frac{1}{2}$ tooth to give $\frac{1}{2}$ tooth advance.

3. Rotate gear clockwise 3 spline keys and pinion clockwise $\frac{3}{4}$ tooth to give $\frac{3}{4}$ tooth advance.

4. Rotate gear clockwise 4 splines and pinion clockwise 1 tooth to give 1 tooth advance.

5. Rotate gear clockwise 2 splines and pinion clockwise $\frac{1}{2}$ tooth to give $\frac{1}{2}$ tooth advance. Now rotate pinion counterclockwise one spline to give $\frac{1}{10}$ tooth advance.

6. Rotate gear counterclockwise one spline and pinion counterclockwise $\frac{1}{4}$ tooth to give $\frac{1}{4}$ tooth retard. Now rotate pinion clockwise one

spline to give $\frac{3}{20}$ tooth advance.

By reversing each step of the above, we may obtain exactly the same values for retard. It is seen that we have a range of adjustment, either advance or retard, from a whole tooth to one tenth tooth of the camshaft gear. Of course, this must be multiplied by 2 to find actual crankshaft degrees.

(I.) General Crankcase Design.

The crankcase does not lend itself well to mathematical treatment, but must be designed by the use of good engineering judgement. The chief stresses to which a crankcase is subjected are:

1. Stress in crankcase bosses due to explosion reaction.

2. Stress in supporting arms and parts intermediate between the cylinder flanges and arms due to torque reaction.

3. Stress in arms and entire case due to weaving of the frame.

The thickness of the metal (aluminum) around the cylinder openings (according to Heldt) was made $\frac{1}{8}$ inch per inch of cylinder bore or $\frac{5}{8}$ inch. The average thickness of the case is $\frac{1}{4}$ inch. The case was reenforced with $\frac{3}{8}$ inch webs on the inside.

(J.) Oil Pump Design.

The oil pump is of the gear-force feed type. Heldt gives the rule to supply 25 cubic inches of oil per minute per square inch of projected crankshaft bearing surface at the maximum speed of the motor.

The projected area for this engine is 68 sq. inches including crankpins. Therefore the pressure pump should move 68 X 25 = 1700 cu. inches of oil per minute at a crankshaft speed of 1750 R.P.M.

Let

d = pitch diameter of one of the pump gears.

P = diametral pitch.

h = height of teeth = $\frac{2}{P}$

f = width of face of gears.

n = revolutions per minute of gears.

Q = quantity of oil delivered in cubic inches per minute.

Then according to Heldt,

$$Q = \frac{2\pi d\ f\ n}{P}$$

and therefore,

$$P = \frac{2\pi d\ f\ n}{Q}$$

Assume that $d = 1\frac{1}{4}$ inches and that $f = \frac{3}{4}$ inch for the feed gears. Since the oil pump runs at 1.5 engine speed it makes 2625 turns per minute. Therefore:

$$P = \frac{2\pi \times 1.25 \times .75 \times 2625}{1700} = 9.08$$

P was made 8 so that the number of teeth was 1.25 X 8 = 10. The oil return gears were made identical to the pressure gears except the width of face f was made 1 inch.

(K.) Water Pump.

The water pump is of the Liberty type with one outlet. The diameter of the rotor was found from, $D = .85\sqrt[4]{V}$

Where,

V = total piston displacement of engine in cubic inches.

Therefore,

$$D = .85\sqrt[4]{19.635 \times 7 \times 6} = 4.55 \text{ inches.}$$

Call it $4\frac{1}{2}$ inches.

CONCLUSION.

In concluding the present work the writers wish to emphasize both the special features of construction and the criticism of their results. The salient features of design are stated briefly as follows:

(1) Hollow-stem valves with adjustable tappets, which provides for the direct action of the cam upon the tappet, thereby eliminating the use of rocker arms, push-rods etc.

(2) Overhead hollow camshaft supported by four bronze bushings driven from the crankshaft by means of a vertical shaft and bevel gears.

(3) Lower vertical drive provided with ball bearings for the reduction of friction.

(4) Magneto bases cast integral with upper crankcase allowing compact construction.

(5) Light cylinder construction made possible through the use of an aluminum jacket and steel sleeves.

(6) Aluminum alloy pistons using spring rings to lock piston pin.

(7) Tubular connecting rods allowing increased strength with reduction in weight.

(8) Seven bearing hollow crankshaft.

(9) Aluminum alloy crankcase with the main crankshaft bearings supported in both the upper and lower halves. Long bolts passing through both halves of crankcase. Cylinders held in position by triangular bridge clamps, a design which adds to the stiffness of the crankcase construction.

(10) Forced feed lubrication using the dry sump proposition.

(11) Two "Dixie" magnetos firing two plugs per cylinder a prevention against misfiring due to fouled spark plugs. The following are the criticisms offered by the writers:

(1) Owing to the accurate machine work which must be performed on the cylinder, the cost of production is against this particular design. As a

commercial engine, therefore, the cylinder design would not be favored.

(2) The crankshaft main bearings cannot be adjusted, a point contrary to the design. As is seen from the drawings that should a bushing become worn it must be replaced with a new one. The joint between the two halves of the case must be lapped using this construction - an expensive operation. However, the Liberty and Hispano-Suiza designs favor this construction. The oil film is not broken and this, no doubt is the reason for its use in the above engines.

(3) As is previously stated, the propeller hub was taken from the United States Army Aviation Standards. This hub appears to be rather heavy and by extended study could, no doubt, be reduced in weight.

The following suggestions may be considered should the motor be redesigned:

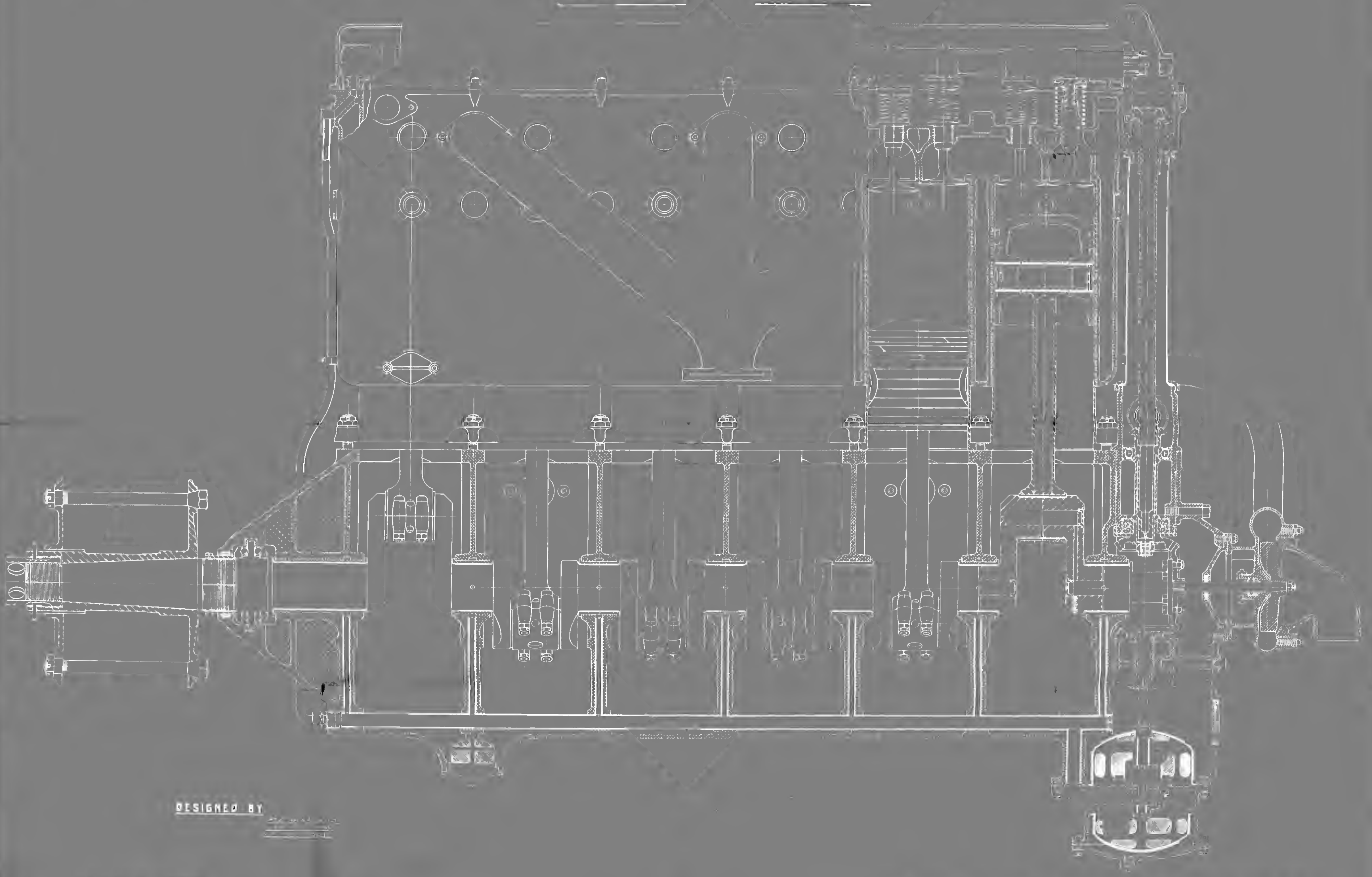
(1) Integral construction of aluminum cylinder block and upper crankcase, the steel cylinder liners

being pressed into place. This construction calls for a detachable cylinder head with cast iron valve seats cast in the head. Using this construction of cylinder and crankcase, the steel sleeves may be obviated by electroplating a thin coat of iron directly on the aluminum wall. This construction has been used on a small air cooled motor and is a matter of future development.

(2) The use of steel crankshaft bearing caps in place of the construction used in the present design.

(3) The use of sliding vane oil pumps.

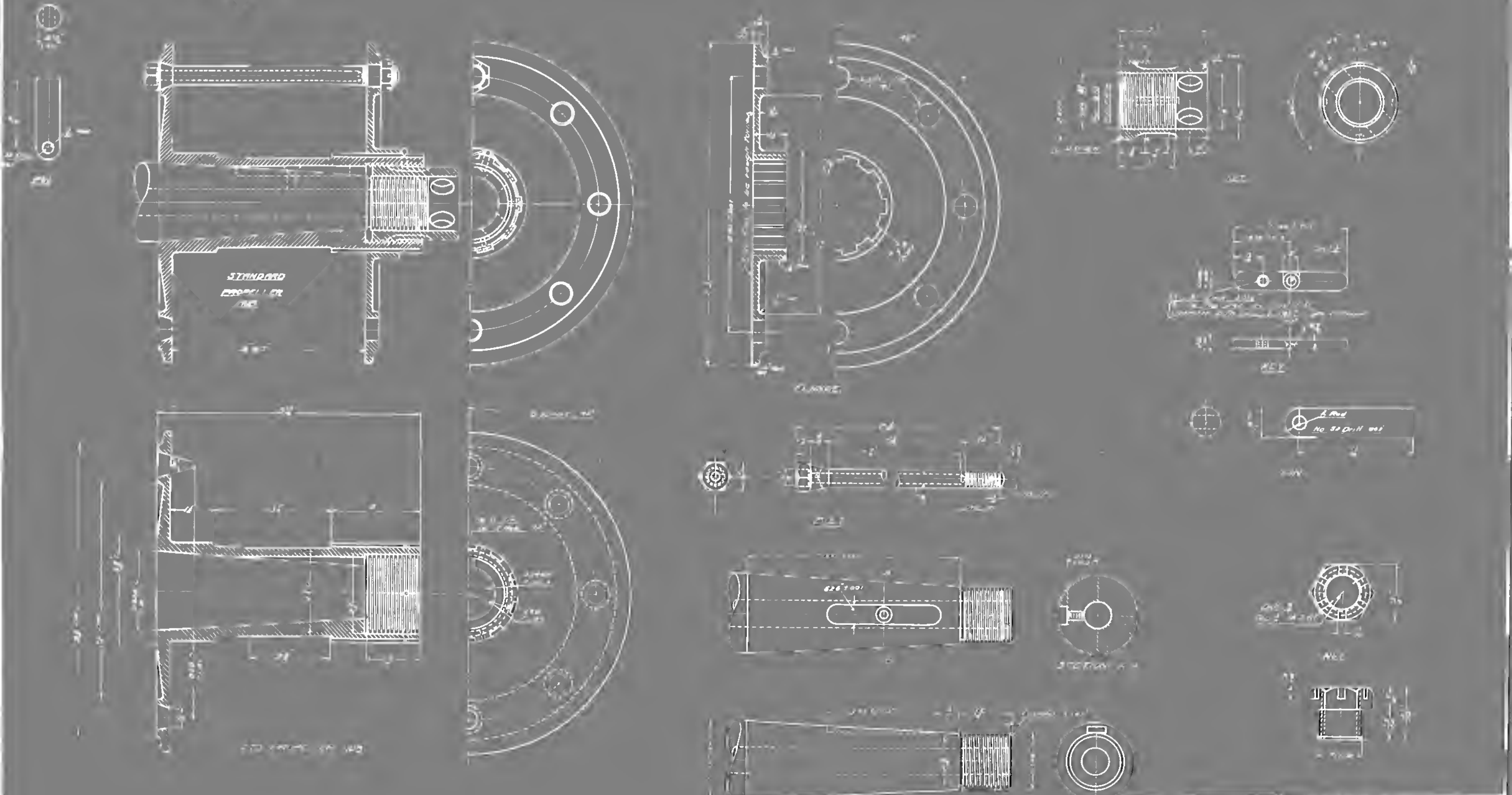
SIX CYLINDER MONO-BLOCK AVIATION ENGINE.



DESIGNED BY

THESIS-1920

HORSEPOWER RATING- 200 AT 1750 R.P.M.


STANDARD
PROPELLER
HUB
FLANGE
KEY
SECTION A-A
No. 30 Drill

DESIGN OF
200 H.P. AERONAUTICAL ENGINE

SCALE HALF FULL SIZE

DESIGNERS

[illegible]

[illegible]

[illegible]

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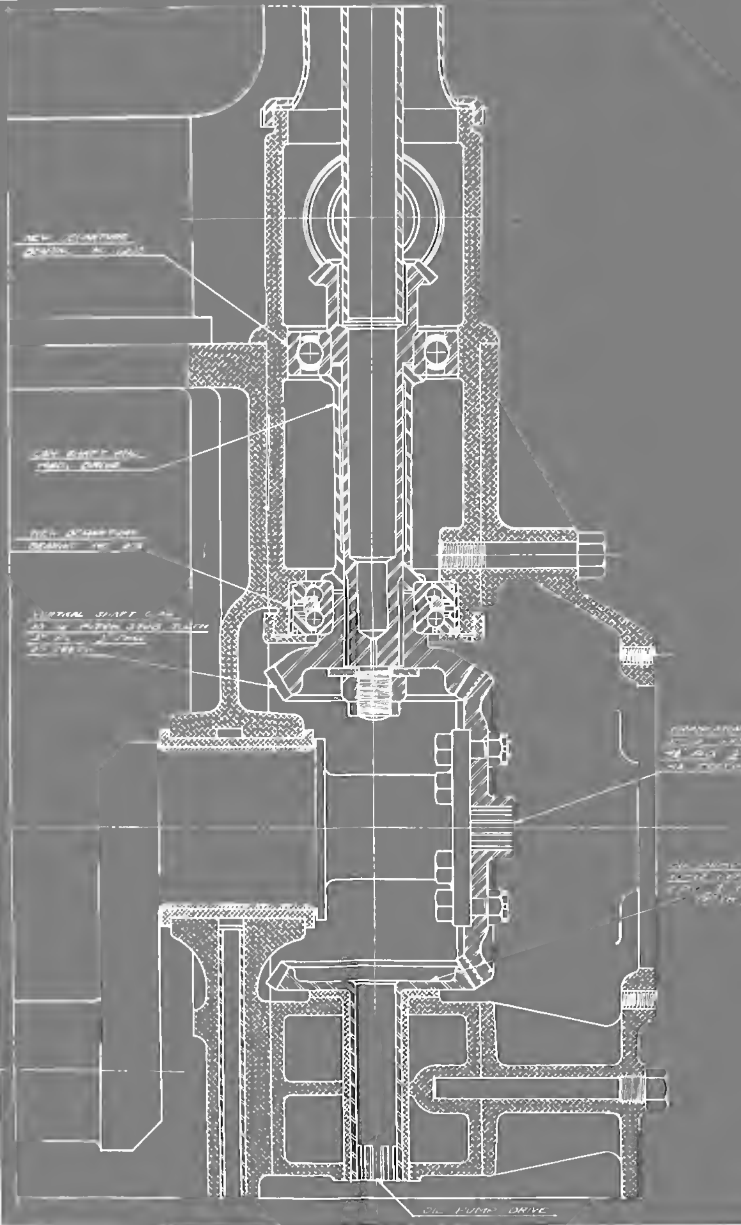
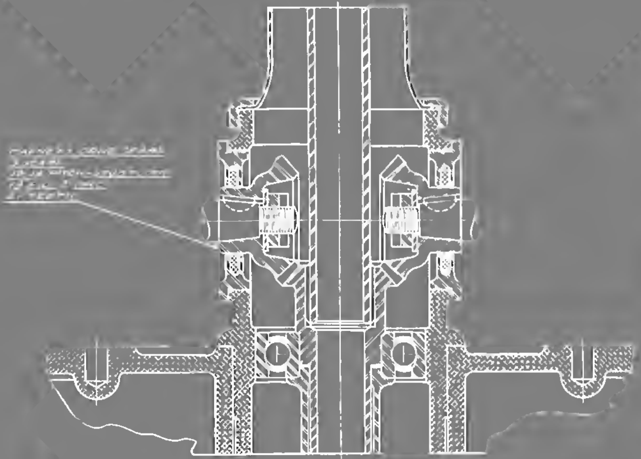
THESIS [illegible]

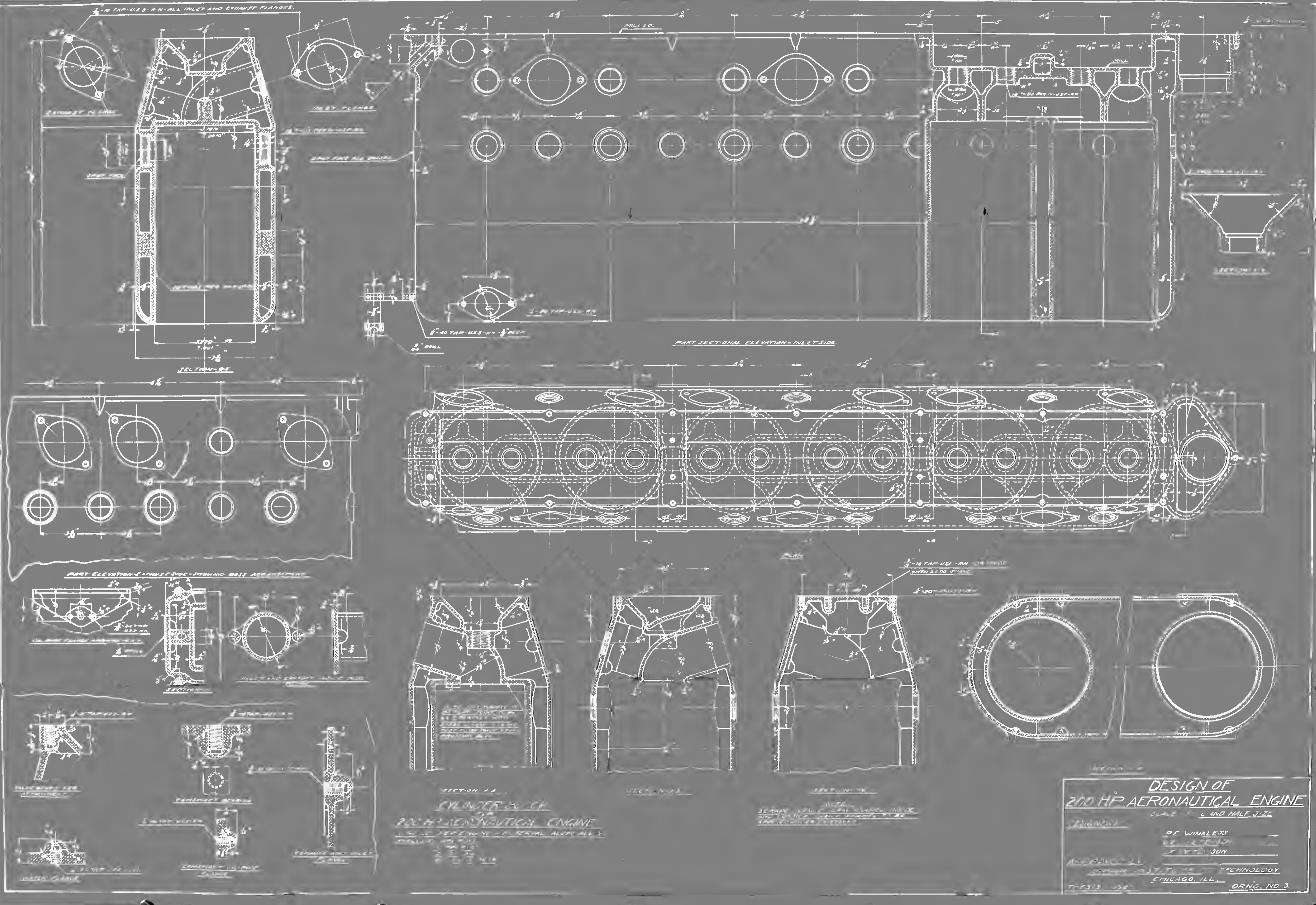
DRAWING NO. 1





GENERAL LAYOUT
LOWER VERTICAL DRIVE SHAFT
FOR
200 H.P. AERONAUTICAL ENGINE

DESIGN OF
200 H.P. AERONAUTICAL ENGINE
SCALE FULL SIZE.
DESIGNERS:
[illegible]
[illegible]
[illegible]
APPROVED BY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.
THESIS [illegible]
DRAWING. NO. 2


$\frac{5}{16}$-18 TAP-U.S.S.-R.H.-ALL INLET AND EXHAUST FLANGES.
EXHAUST FLANGE
INLET FLANGE
MILL ED.
SECTION-BB
PART SECTIONAL ELEVATION - INLET SIDE.
PART ELEVATION - EXHAUST SIDE - SHOWING BOSS ARRANGEMENT
PLAN
SECTION 2-2
SECTION-AA
WATER FLANGE
MILL

CYLINDER BLOCK
FOR
200 H.P. AERONAUTICAL ENGINE

DESIGN OF
200 H.P. AERONAUTICAL ENGINE

SCALE FULL AND HALF SIZE

DESIGNED BY

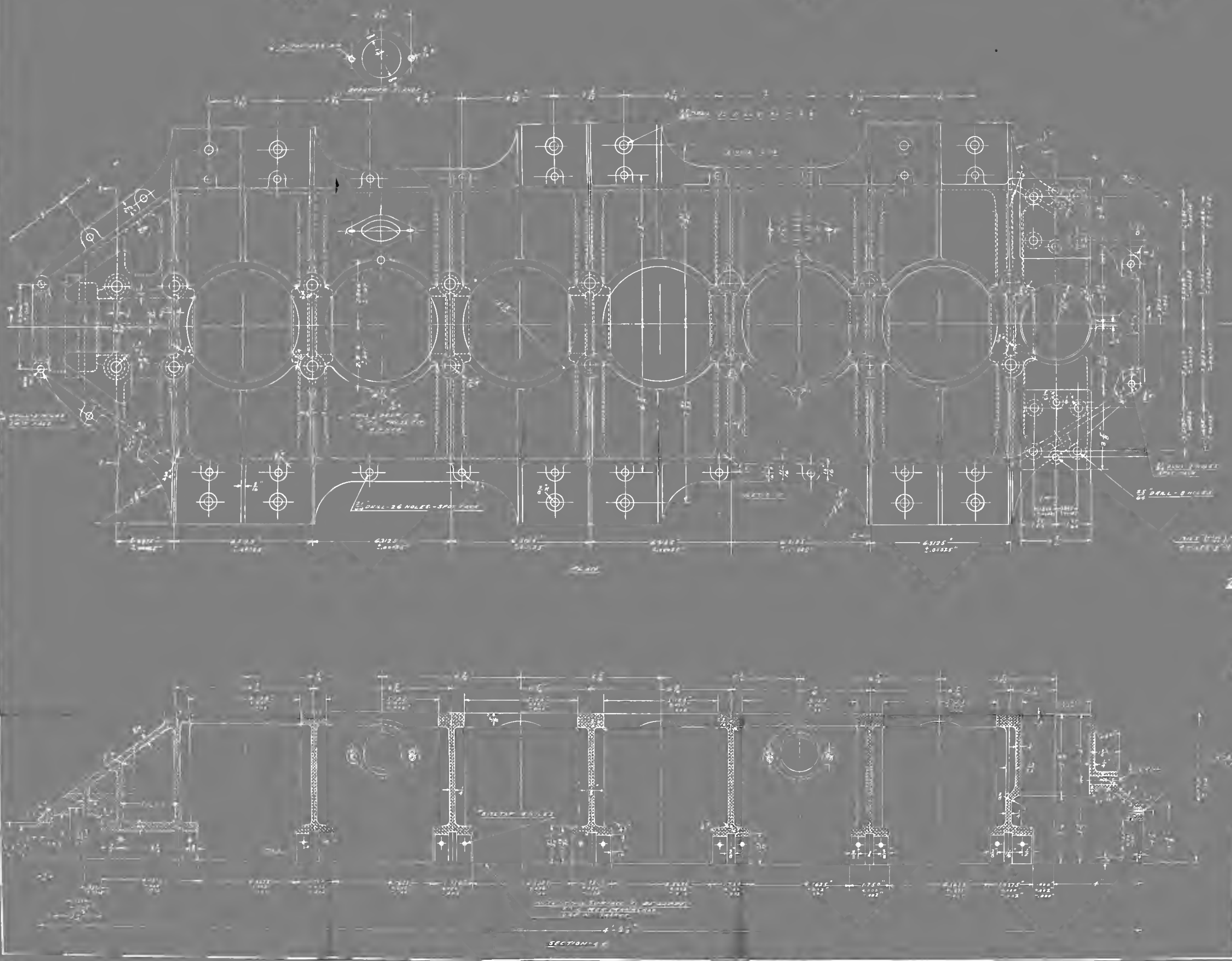
P.E. WINKLESS
[illegible]
[illegible]SON

APPROVED [illegible]

ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

THESIS [illegible]

DRNG. NO. 3.


BREATHER FLANGE
21/64 DRILL-26 HOLES-SPOT FACE
25/64" DRILL - 8 HOLES
6.3125"
6.3125
PLAN
SECTION-A-B
4'-8 3/4"

UPPER CRANKCASE

FOR

200 H.P. AERONAUTICAL ENGINE

[illegible]

ALUMINUM - 87.0

COPPER - [illegible]

[illegible]

[illegible]

DESIGN OF

200 H.P. AERONAUTICAL ENGINE

SCALE - HALF SIZE.

DESIGNERS

[illegible]LESS.

C. O. PETERSON

[illegible] PETERSON.

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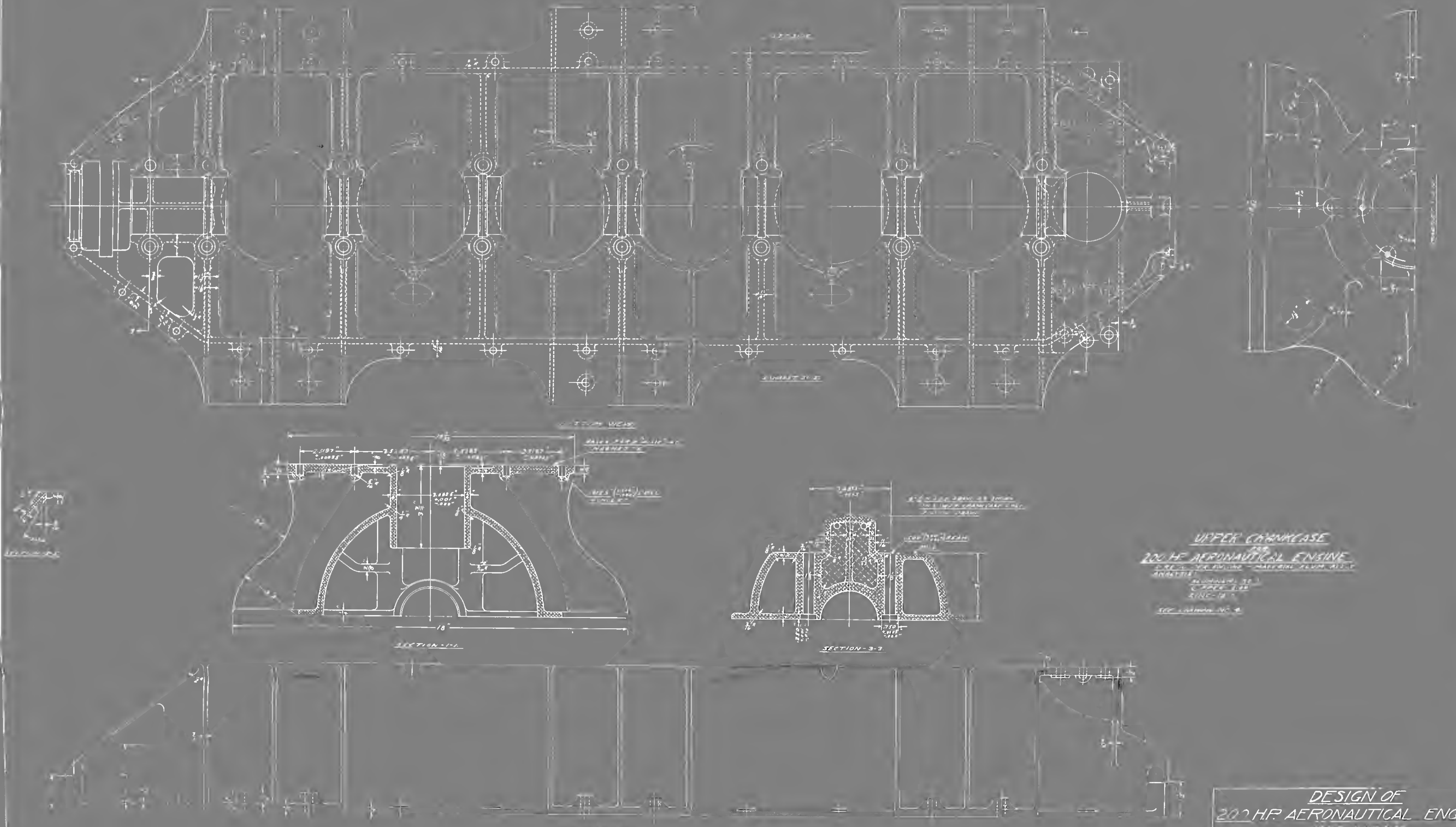
CHICAGO, ILL.

THESIS - 192[illegible]

DRNG. NO. 4.


SECTION-1-1
SECTION-3-3
SIDE ELEVATION
18"

UPPER CRANKCASE
FOR
200 H.P. AERONAUTICAL ENGINE

[illegible] – MATERIAL ALUM. ALLOY
ANALYSIS:
ALUMINUM [illegible]
COPPER [illegible]
ZINC [illegible]

SEE [illegible] NO. 4

DESIGN OF
200 H.P. AERONAUTICAL ENGINE

SCALE [illegible]

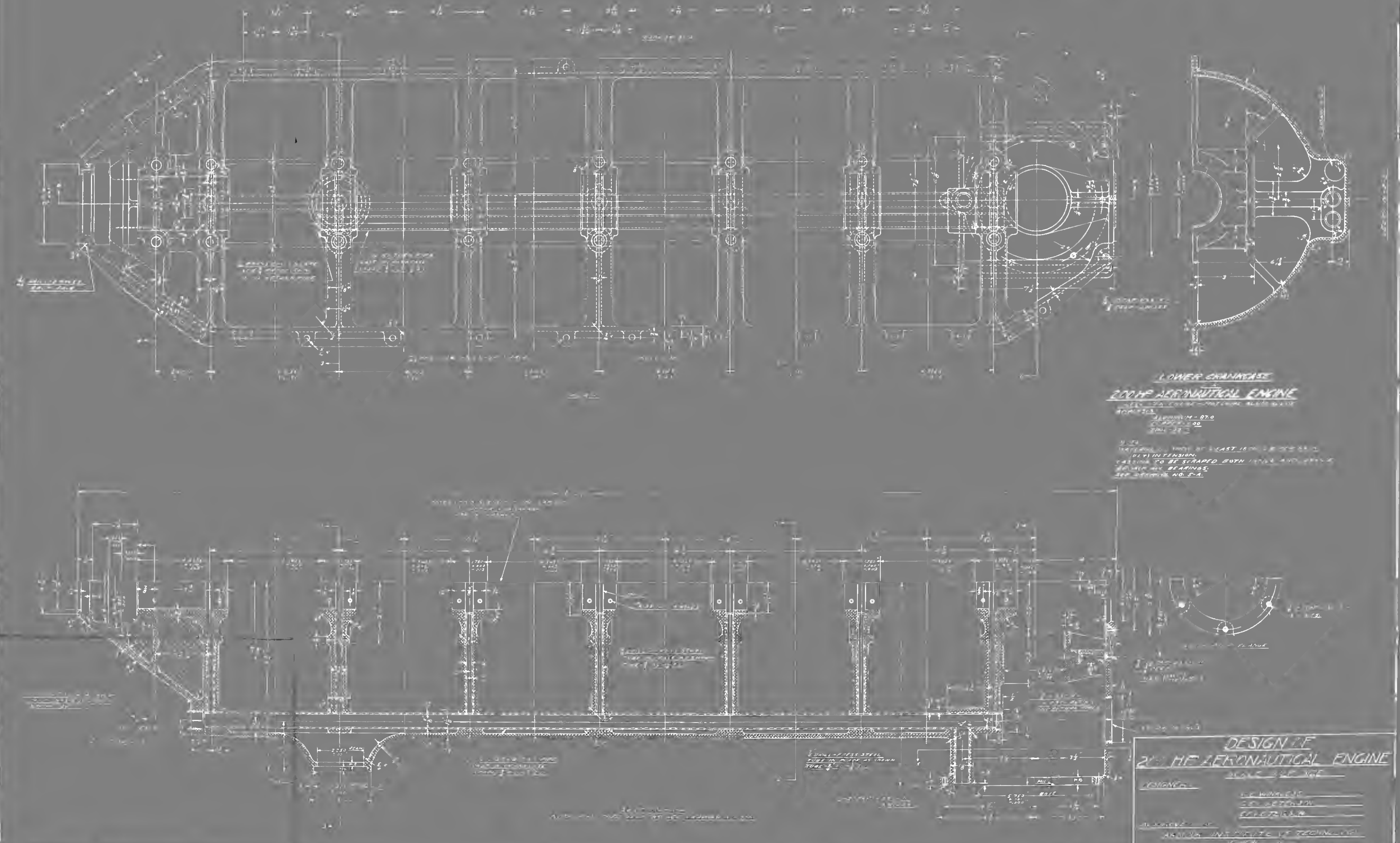
DESIGNED [illegible]

[illegible]

ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO ILL.

THESIS [illegible]

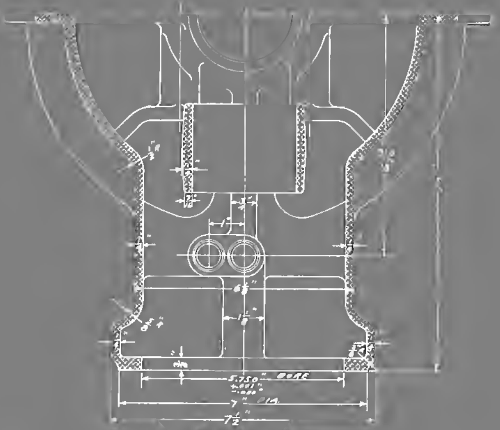
SHEET NO. 4



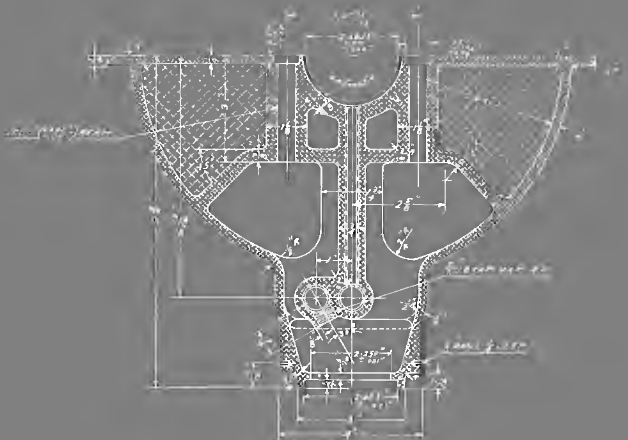
LOWER CRANKCASE
200 HP AERONAUTICAL ENGINE
ALUMINUM - 87.0
00
ULT) IN TENSION.
CASTING TO BE SCRAPED BOTH
BEARINGS.
SEE DRAWING NO. 5-A.

DESIGN OF
200 HP AERONAUTICAL ENGINE

ARMOUR INSTITUTE OF TECHNOLOGY


5.750" BORE
7" DIA
7½"
6⅝"
1½"

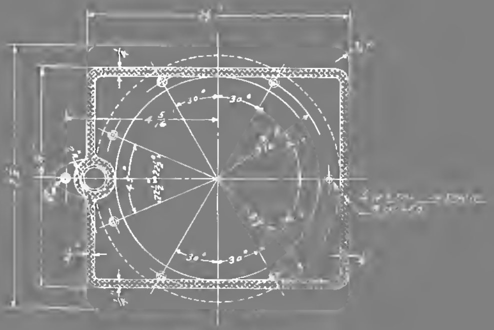
SECTION-1-1



SECTION-2-2

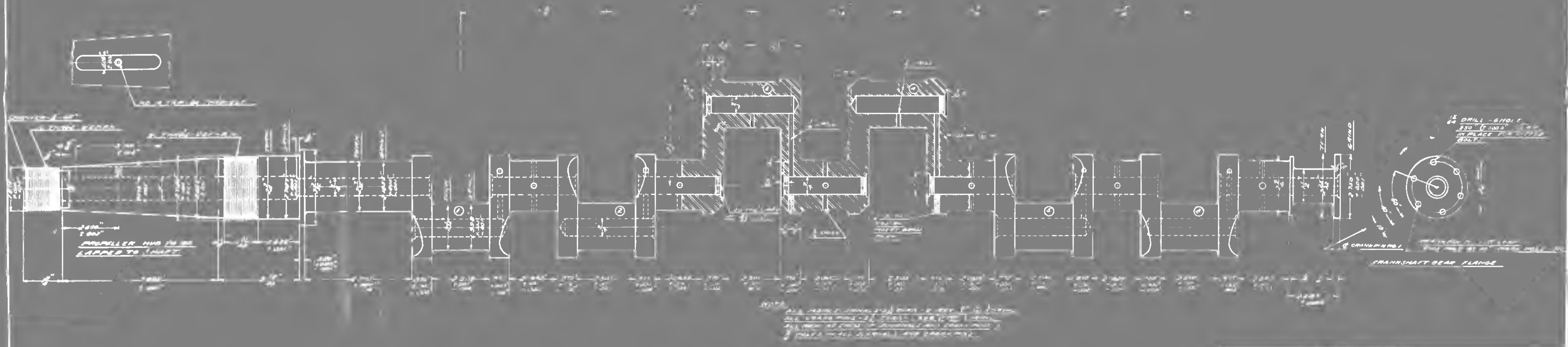
LOWER CRANKCASE
OF
200 HP AERONAUTICAL ENGINE.

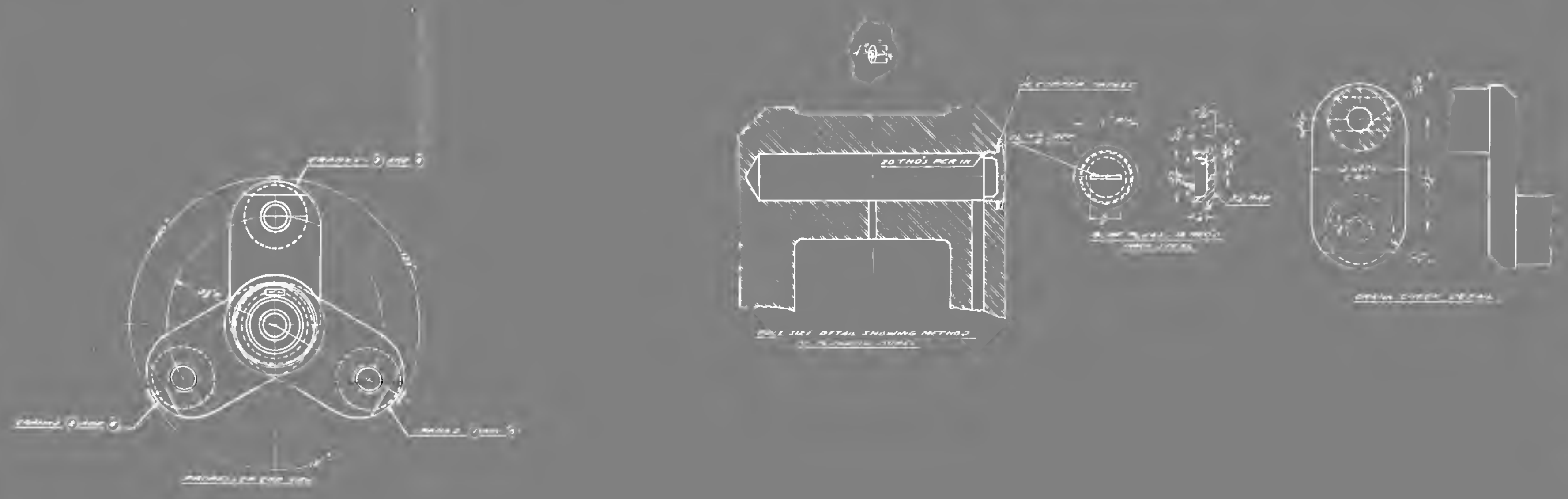
(8 CYLINDER ENGINE – MATERIAL ALUMINUM ALLOY
ANALYSIS:
ALUMINUM – [illegible]
COPPER – [illegible]
ZINC – [illegible]
SEE DRAWING NO. 5


30°
30°
30°
30°
4 5/16"

SECTION [illegible]

DESIGN OF
200 HP AERONAUTICAL ENGINE
SCALE HALF SIZE
DESIGNED BY:
[illegible]
[illegible]
[illegible]
APPROVED BY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO ILL.
[illegible]
DWG. No. [illegible]


PROPELLER HUB TO BE
LAPPED TO SHAFT
GRIND
CRANKSHAFT GEAR FLANGE


COPPER WASHER
20 THD'S PER IN
FULL SIZE DETAIL SHOWING METHOD
OF PLUGGING HOLES
CRANK CHEEK DETAIL
PROPELLER END VIEW

CRANKSHAFT
FOR
200 H.P. AERONAUTICAL ENGINE

[illegible]

... READING TO BE TAKEN AFTER ...

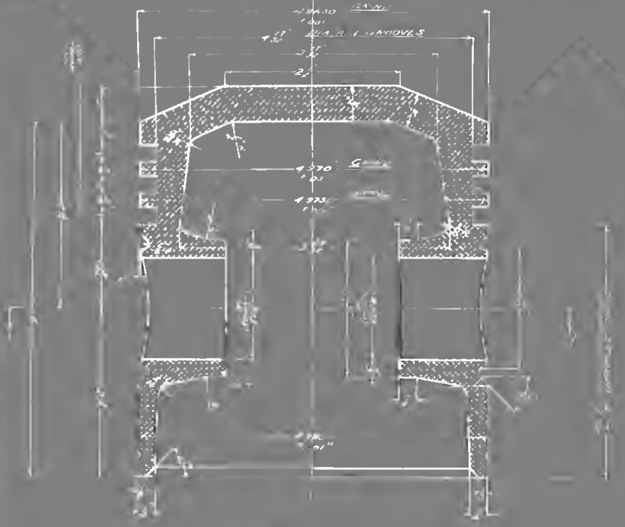
[illegible]

DESIGN OF
200 H.P. AERONAUTICAL ENGINE.
SCALE HALF SIZE.

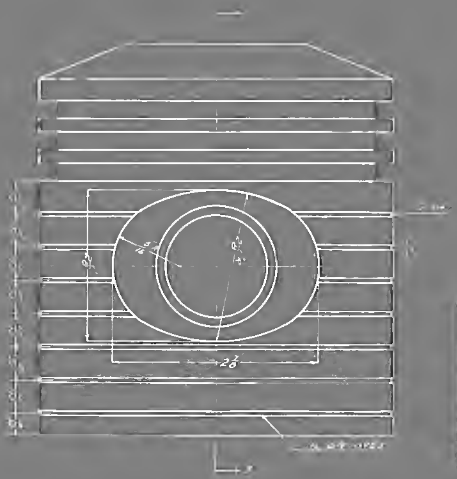
DESIGNED BY
... WINKLESS
C. O. PETERSON
... PETERSON

... TECHNOLOGY.
CHICAGO, ILL.

DRNG. NO. 6.


GAUGE
DIA. 4 GROOVES
GAUGE
GAUGE

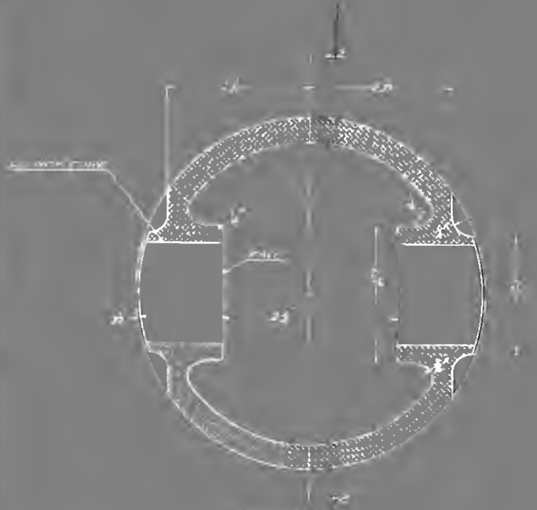
SECTION 1-1


$2\frac{7}{8}$
4 GROOVES

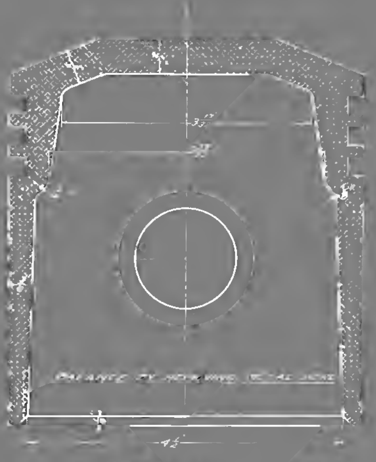
PISTON
FOR
200 H.P. AERONAUTICAL ENGINE.

6 REQ'D - MATERIAL [illegible]
ANALYSIS
ALUMINUM [illegible]
[illegible]

PISTONS WH[illegible]
WITHIN $\frac{1}{4}$ OZ.



[illegible]


BALANCE BY REMOVING METAL HERE
$4\frac{1}{2}$

SECTION 2-2

DESIGN OF
200 H.P. AERONAUTICAL ENGINE

SCALE [illegible]

DESIGNED

[illegible]
[illegible]
[illegible]

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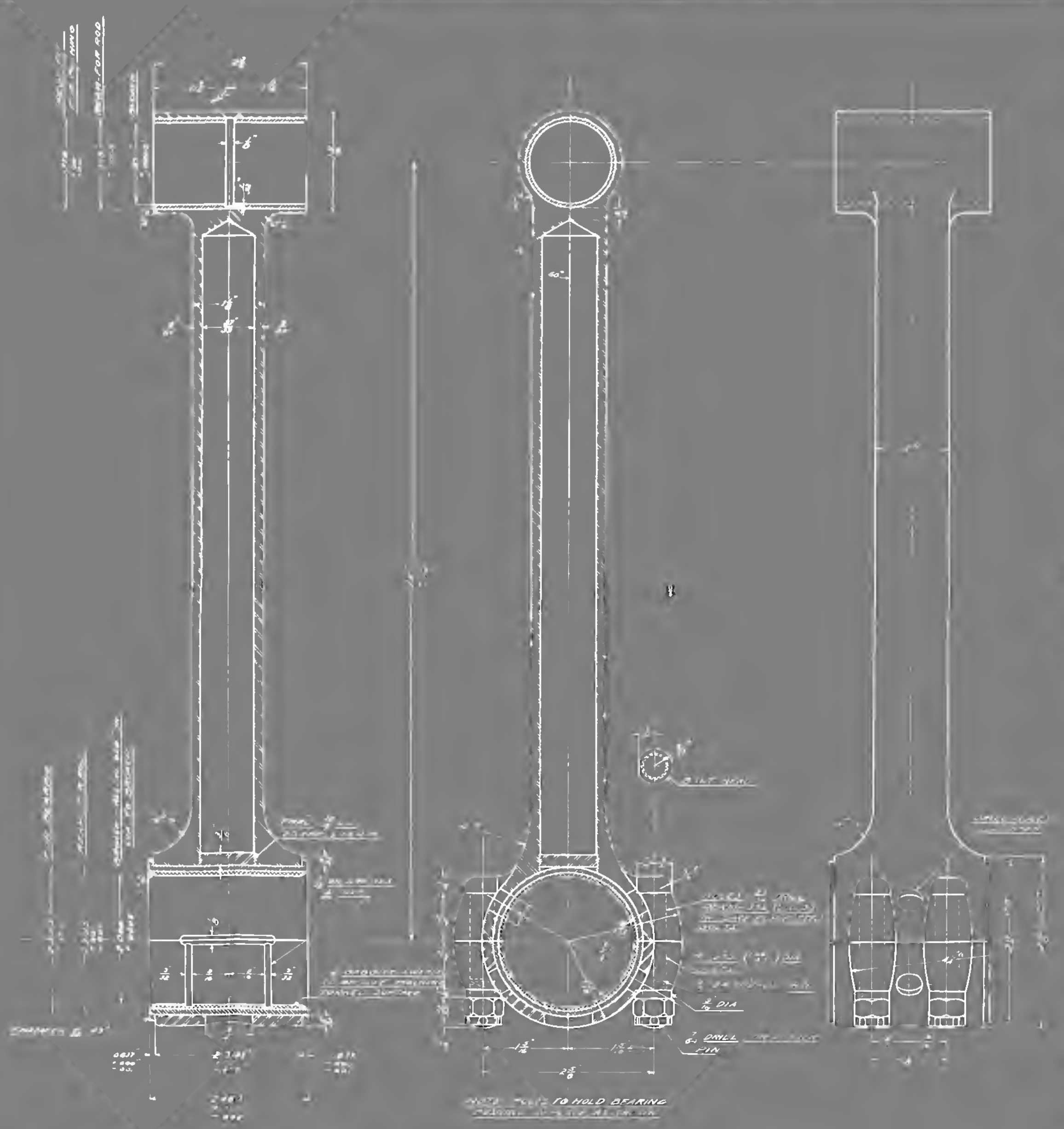
THESIS - 192[illegible]

DRAWING NO. [illegible]


NOTE: BOLTS TO HOLD BEARING
1 5/16
1 5/16
2 5/8
5/16 DIA
DRILL
PIN

CONNECTING ROD
FOR
200 H.P. AERONAUTICAL ENGINE

CHROME NICKEL STEEL - [illegible]

ANALYSIS [illegible]

HEAT TREATMENT BEFORE MACHINING

CLEARANCES

CONNECTING ROD BOLTS [illegible] STEEL

ANALYSIS: [illegible]

HEAT TREATMENT [illegible]

BRONZE ANALYSIS [illegible]

BABBITT ANALYSIS [illegible]

DESIGN OF
200 H.P. AERONAUTICAL ENGINE
SCALE: FULL SIZE

DESIGNERS:
H. E. WINKLESS
O. C. O. PETERSON
C. F. PETERSON

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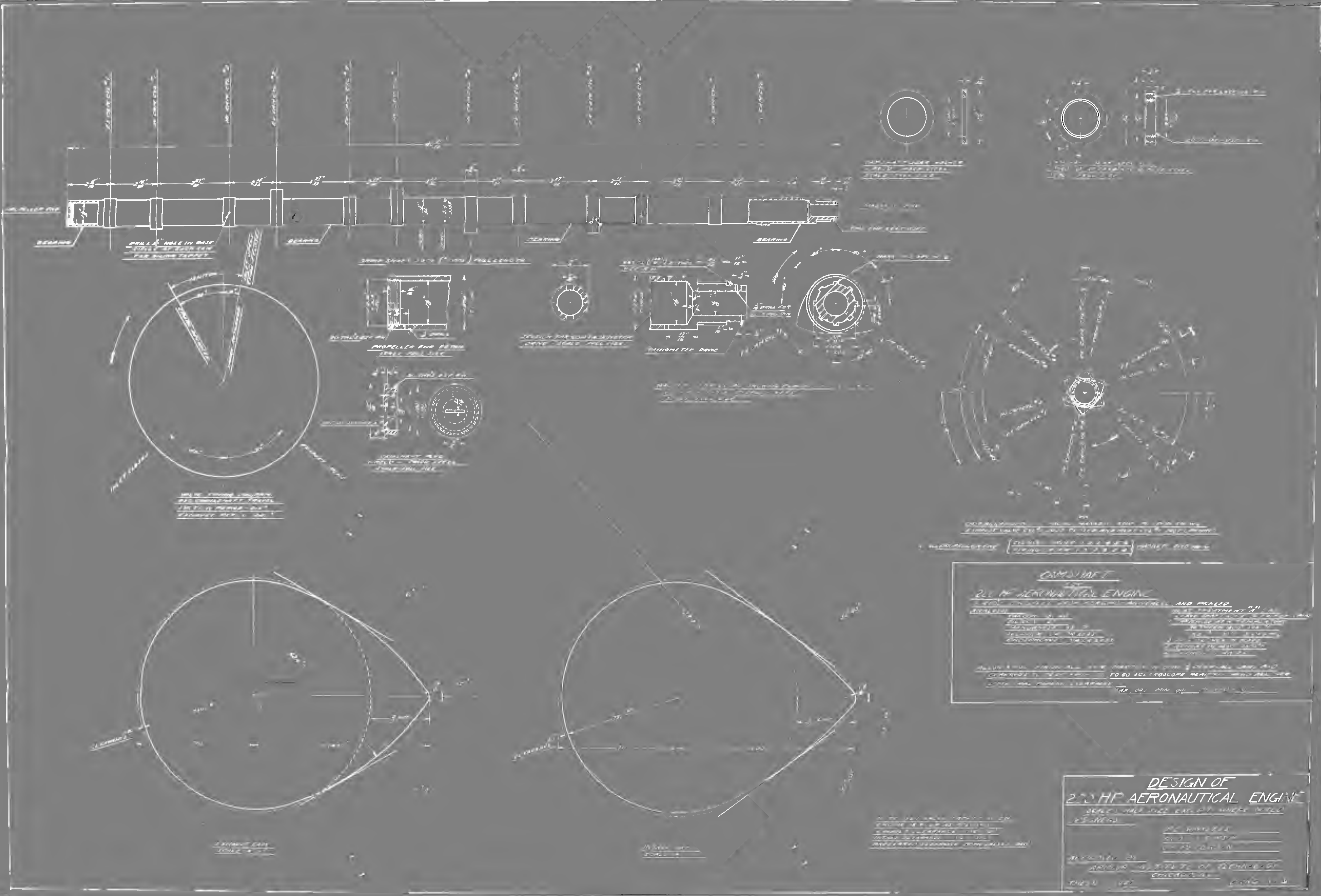
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

THESIS - 1920
DRAWING NO. 8.


BEARING
DRILL 3/8" HOLE IN BASE
FOR ... TAPPET
BEARING
BEARING
BEARING
PROPELLER END DETAIL
SCALE FULL SIZE
TACHOMETER DRIVE
EXHAUST CAM
INTAKE CAM

CAMSHAFT

FOR

2[illegible] HP AERONAUTICAL ENGINE

AND PICKLED

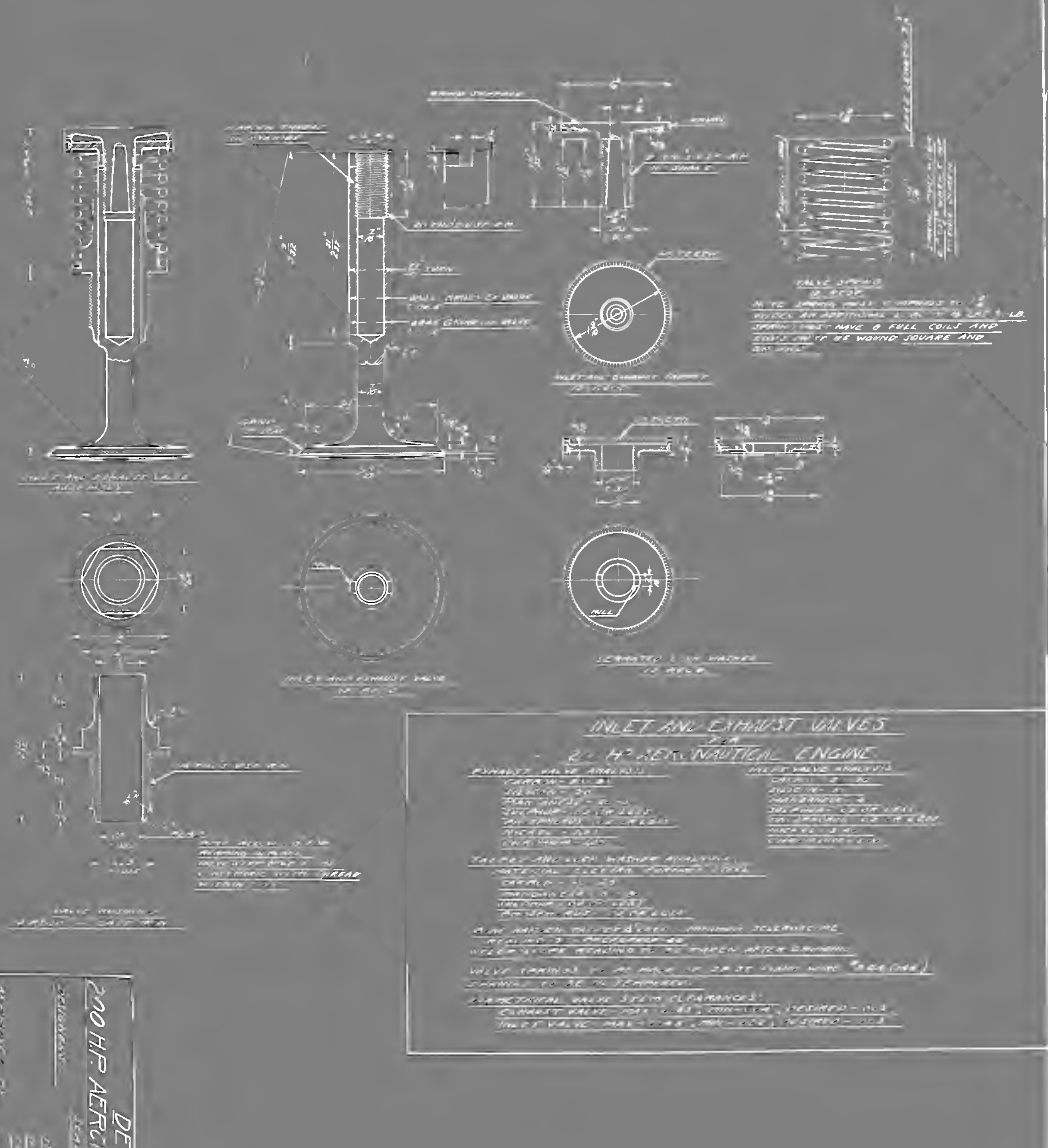
HEAT TREATMENT "A"

TO 80 SCLEROSCOPE HEA

MAX .003 MIN .001"

DESIGN OF

200 HP AERONAUTICAL ENGINE


VALVE SPRING
12 REQ.
... HAVE 8 FULL COILS AND
... BE WOUND SQUARE AND
SERRATED LOCK WASHER
12 REQ.
INLET AND EXHAUST TAPPET
INLET AND EXHAUST VALVE
THREAD
FULL

INLET AND EXHAUST VALVES
FOR
200 H.P. AERONAUTICAL ENGINE

EXHAUST VALVE ANALYSIS [illegible]

INLET VALVE ANALYSIS [illegible]

TAPPET AND LOCK WASHER ANALYSIS [illegible]

SYMMETRICAL VALVE STEM CLEARANCES:
EXHAUST VALVE – MAX [illegible], MIN [illegible], DESIRED [illegible]
INLET VALVE – MAX [illegible], MIN [illegible], DESIRED [illegible]

DESIGN OF
200 H.P. AERONAUTICAL ENGINE
SCALE: FULL SIZE
DESIGNERS:
[illegible]
APPROVED BY
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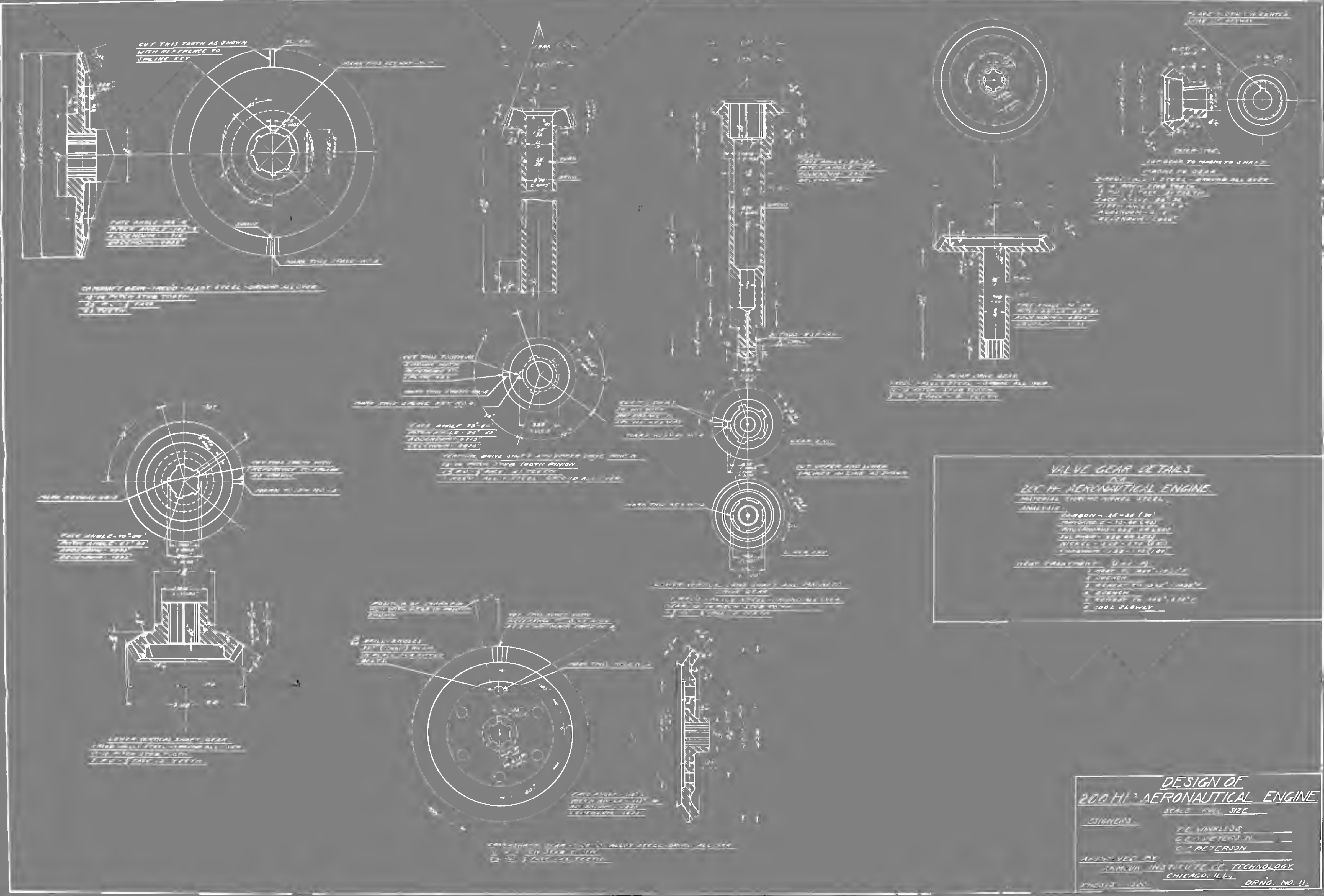
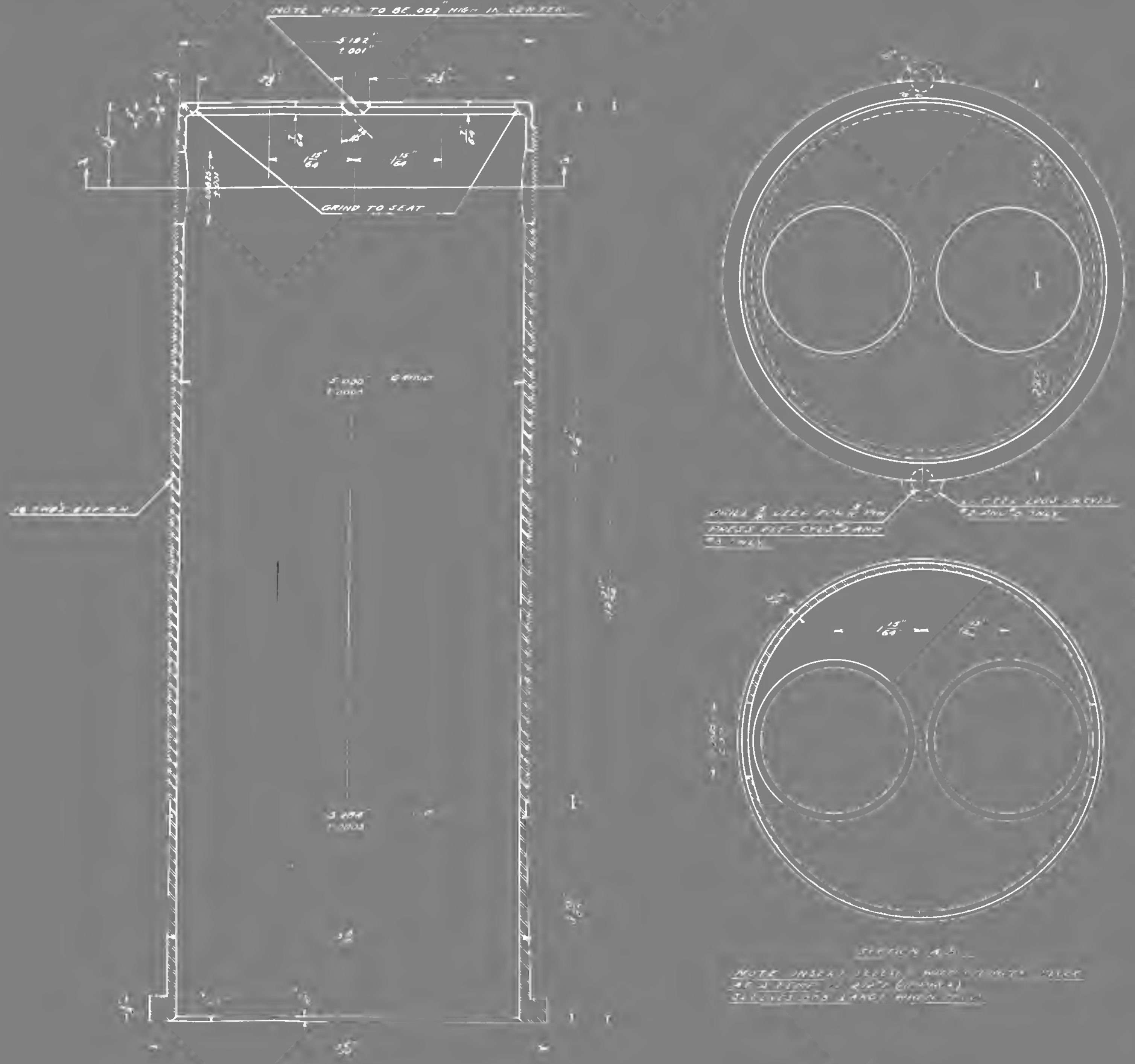
CHICAGO, ILL.
THESIS 1920
DWG. NO. [illegible]


CUT THIS TOOTH AS SHOWN WITH REFERENCE TO SPLINE KEY
CUT THIS TOOTH AS SHOWN WITH REFERENCE TO SPLINE KEY
FACE ANGLE 53°-50'
PITCH ANGLE - 36°-52'
ADDENDUM - .0715"
VERTICAL DRIVE SHAFT AND UPPER DRIVE PINION
12 HP PITCH STUB TOOTH PINION
LOWER VERTICAL SHAFT GEAR
CUT GEAR TO MAGNETO SHAFT

VALVE GEAR DETAILS
FOR
200 HP AERONAUTICAL ENGINE.

MATERIAL CHROME NICKEL STEEL
ANALYSIS
CARBON - .25-.35 (.30)
[illegible]
HEAT TREATMENT [illegible]
1 HEAT TO [illegible]
2 QUENCH
3 REHEAT TO [illegible]
4 QUENCH
5 REHEAT TO [illegible]
6 COOL SLOWLY

DESIGN OF
200 HP AERONAUTICAL ENGINE.
SCALE FULL SIZE.
DESIGNERS
[illegible]
[illegible]
[illegible] PETERSON
APPROVED BY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.,
THESIS [illegible]
DRNG. NO. 11.



NOTE HEAD TO BE .002" HIGH IN CENTER
5.192"
1.001"
1 15/64
GRIND TO SEAT
5.000
GRIND
16 THDS PER IN
SECTION A-B
NOTE INSERT SLEEVES [illegible]
AT A TEMP OF [illegible]
SLEEVES AND [illegible]
DRILL 3/8 HOLE FOR 3/8 PIN
PRESS FIT

STEEL SLEEVES
FOR
200 HP AERONAUTICAL ENGINE

[illegible] .30 CARBON STEEL

ANALYSIS

CARBON .25 .35 (.3)
MANGANESE .50 [illegible] (.60)
SULPHUR .04 OR LESS
PHOSPHORUS .04 OR LESS

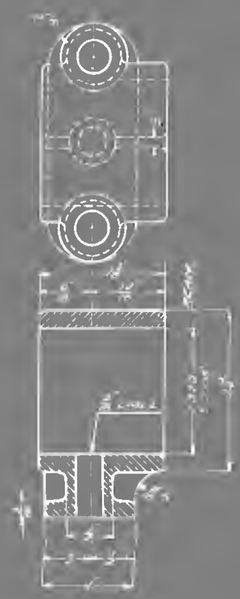
HEAT TREATMENT [illegible]
1 HEAT TO 1600° 1650°F
2 COOL SLOWLY
3 HEAT TO 1450° [illegible]
4 QUENCH
5 REHEAT TO 300 350°F
6 COOL SLOWLY

DESIGN OF
200 H.P. AERONAUTICAL ENGINE
SCALE FULL SIZE

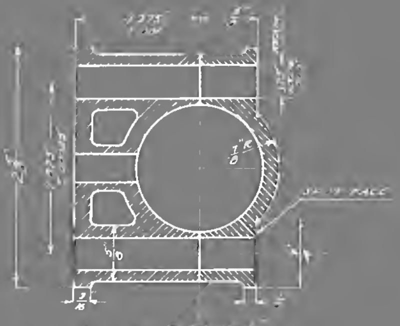
DESIGNERS
F.R. WINKLESS
[illegible] PETERSON
[illegible] PETERSON

APPROVED BY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO ILL.

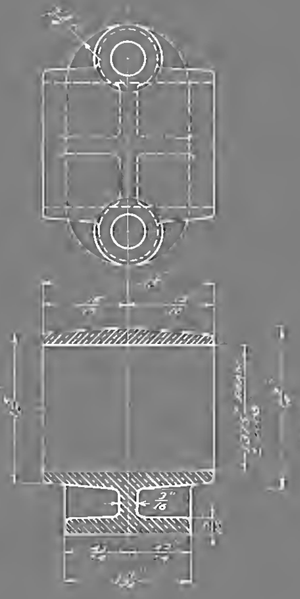
[illegible] 13 1920
DWG. NO. 12



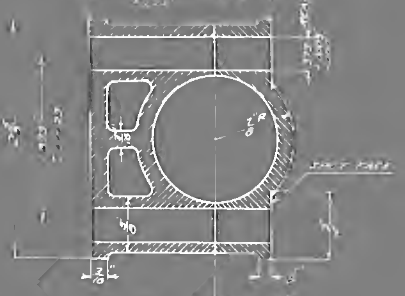
PROPELLER END CAMSHAFT BEARING
1 REQ. — BRONZE DIE CAST
ANALYSIS:
TIN - 90%
ANTIMONY - 8%
COPPER - 2%


1.375
2.001
REAM
1/8 R
SPLIT FACE

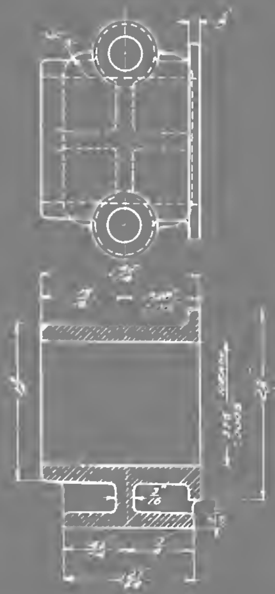
ALL FILLET RADII - 1/8
SPLIT BEARING



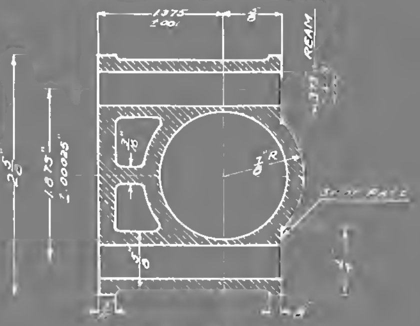
INTERMEDIATE CAMSHAFT BEARING
2 REQ. - BRONZE DIE CAST
ANALYSIS
TIN - 88%
ANTIMONY - 5%
COPPER - 7%


REAM
1/8 R
SPLIT FACE

ALL FILLET RADII - 1/8
SPLIT BEARING



MAGNETO END CAMSHAFT BEARING
1 REQ. BRONZE DIE CAST
ANALYSIS
TIN - 88%
ANTIMONY - 5%
COPPER - 7%


1.375
2.001
REAM
2 5/8"
1.875"
1.0028"
1/8 R
SPLIT FACE

ALL FILLET RADII - 1/8
DO NOT SPLIT THIS BEARING

DESIGN OF
200 H.P. AERONAUTICAL ENGINE

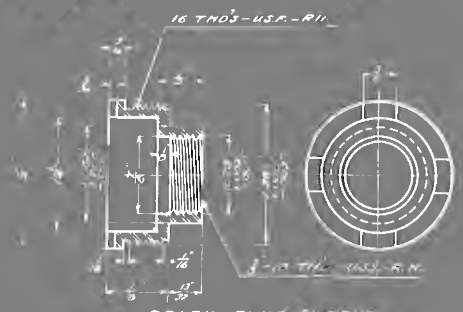
SCALE FULL SIZE

DESIGNERS
P.E. WINKLESS
GEO. PETERSON
C.F. PETERSON

APPROVED BY

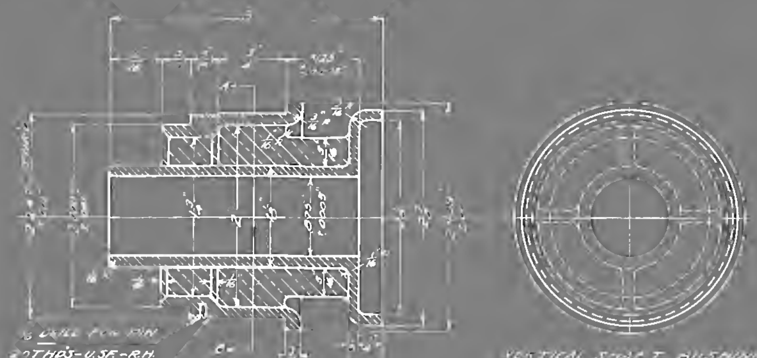
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

THESIS - 1920
DWG. NO. 19

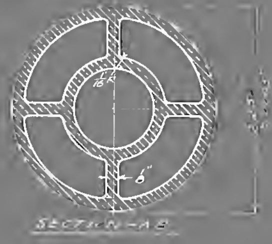

16 THD'S-U.S.F.-R.H.
1/16"
13/32"
THD U.S.S.-R.H.

SPARK PLUG SLEEVE
12 REQ'D. — .30 CARBON STEEL

ANALYSIS:
CARBON — .15 —
MANGANESE — .0 — .4
SULPHUR — .04 OR LESS
PHOSPHORUS — .04 OR LESS

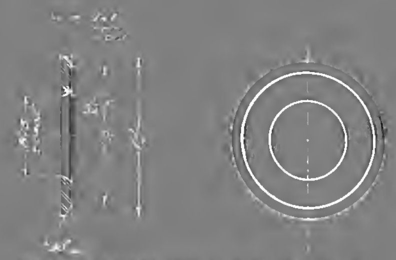

1/8 DRILL FOR PIN
20 THD'S-U.S.F.-R.H.

VERTICAL SHAFT BUSHING
1 REQ'D. — BRONZE CASTING

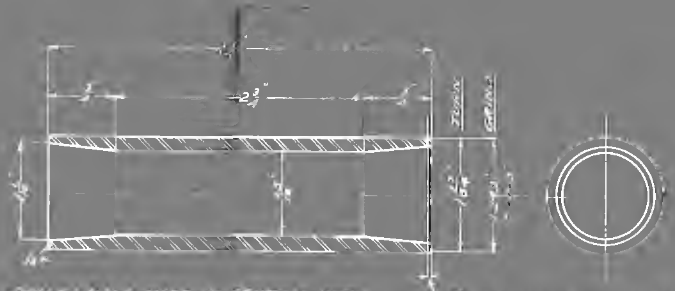


SECTION — A B

ANALYSIS:
TIN — 84%
ANTIMONY — 9%
COPPER — 7%



VERTICAL SHAFT PINION THRUST WASHER
1 REQ'D. — .20-.25 CARBON STEEL — GROUND ALL OVER
CASE HARDEN 1/64 DEEP



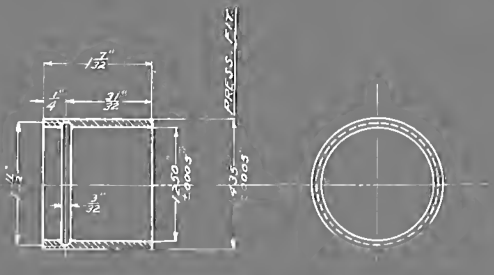
DRILLED HOLE MUST BE CENTRAL WITH OUTSIDE DIA. OF PIN.

1/32 MAX. ALLOWANCE IN EACH END FOR BALANCING

PISTON PIN
6 REQ'D. — CAR. STEEL
CASE HARDEN AND GRIND

ANALYSIS:
CARBON — .20 — .30 (.20)
MANGANESE — .45 — .70 (.48)
SULPHUR — .03 OR LESS
PHOSPHORUS — .03 OR LESS

HARDNESS:
SCLEROSCOPE READING 75 — 85 WITH 80 PREFERRED.
SCLEROSCOPE READING TO BE TAKEN AFTER GRINDING

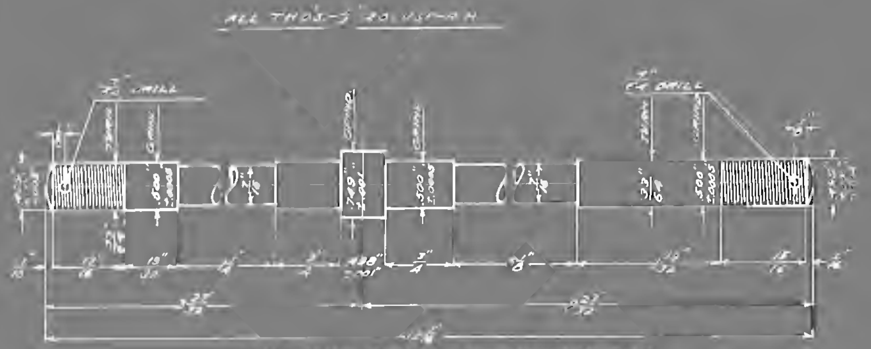

PRESS FIT
1 7/32"
1/4"
31/32"
3/32"

PISTON PIN BUSHING
12 REQ'D. — MATERIAL: BRONZE

ANALYSIS:
TIN — 84%
ANTIMONY — 9%
COPPER — 7%

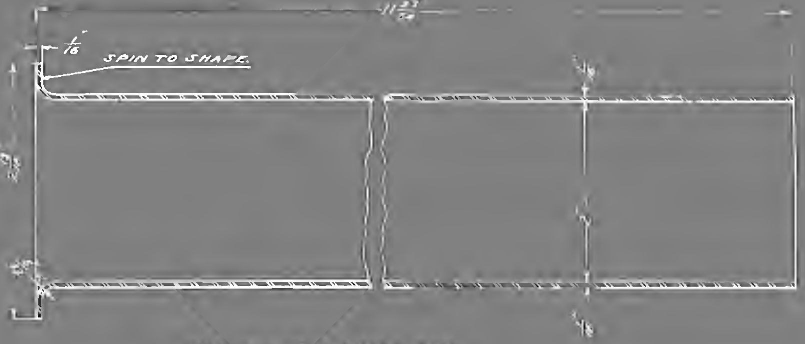
NOTE: BUSHING TO BE DIE CAST

DESIGN OF
200 H.P. AERONAUTICAL ENGINE
SCALE FULL SIZE
DESIGNERS:
R.E. WINKLESS
C.E. PETERSON
C.F. PETERSON
APPROVED BY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO ILL.
THESIS — 1920
DRAWING NO. [illegible]

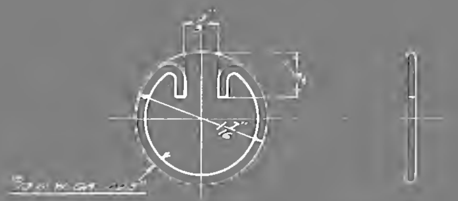

ALL THDS - 5/8" 20 U.S.F. - R.H.
DRILL
64 DRILL
.500 ±.0005
.749 ±.001
.500 ±.0005
.498 ±.001
3/4"
21/32

CRANKSHAFT BEARING AND CYLINDER BOLT.
14 REQD - CHROM. NICKEL STEEL - HEAT TREATED

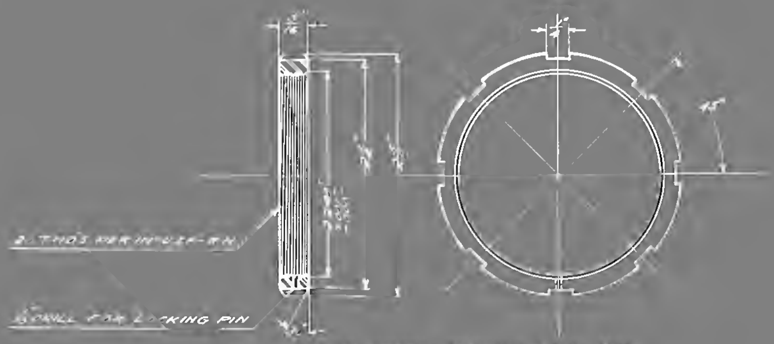
ANALYSIS (S.A.E. NO 3130)
CARBON - .25 - .35
MANGANESE - .50 - .80
PHOSPHORUS - .04 OR LESS
SULPHUR - .04 OR LESS.
NICKEL - 1.00 - 1.50
CHROMIUM - .45 - .75
HEAT TREATMENT (S.A.E. "A")
1 HEAT TO 1500°-1600°F
2 QUENCH.
3 REHEAT TO 600°-1200°F
4 COOL SLOWLY.


SPIN TO SHAPE.
1/16

VERTICAL DRIVE SHAFT CASE
1 REQD. - SEAMLESS STEEL TUBE


.13 B.W. GA. WIRE

PISTON PIN RETAINER SPRING
12 REQD - VANADIUM ALLOY SPRING STEEL (.13 B.W. GA.)
OIL TEMPERED.


2 THDS PER INCH - R.H.
DRILL FOR LOCKING PIN
45°

VERTICAL SHAFT BUSHING LOCK NUT.
1 REQD - [illegible] GA. - C.R. BESSEMER ST.

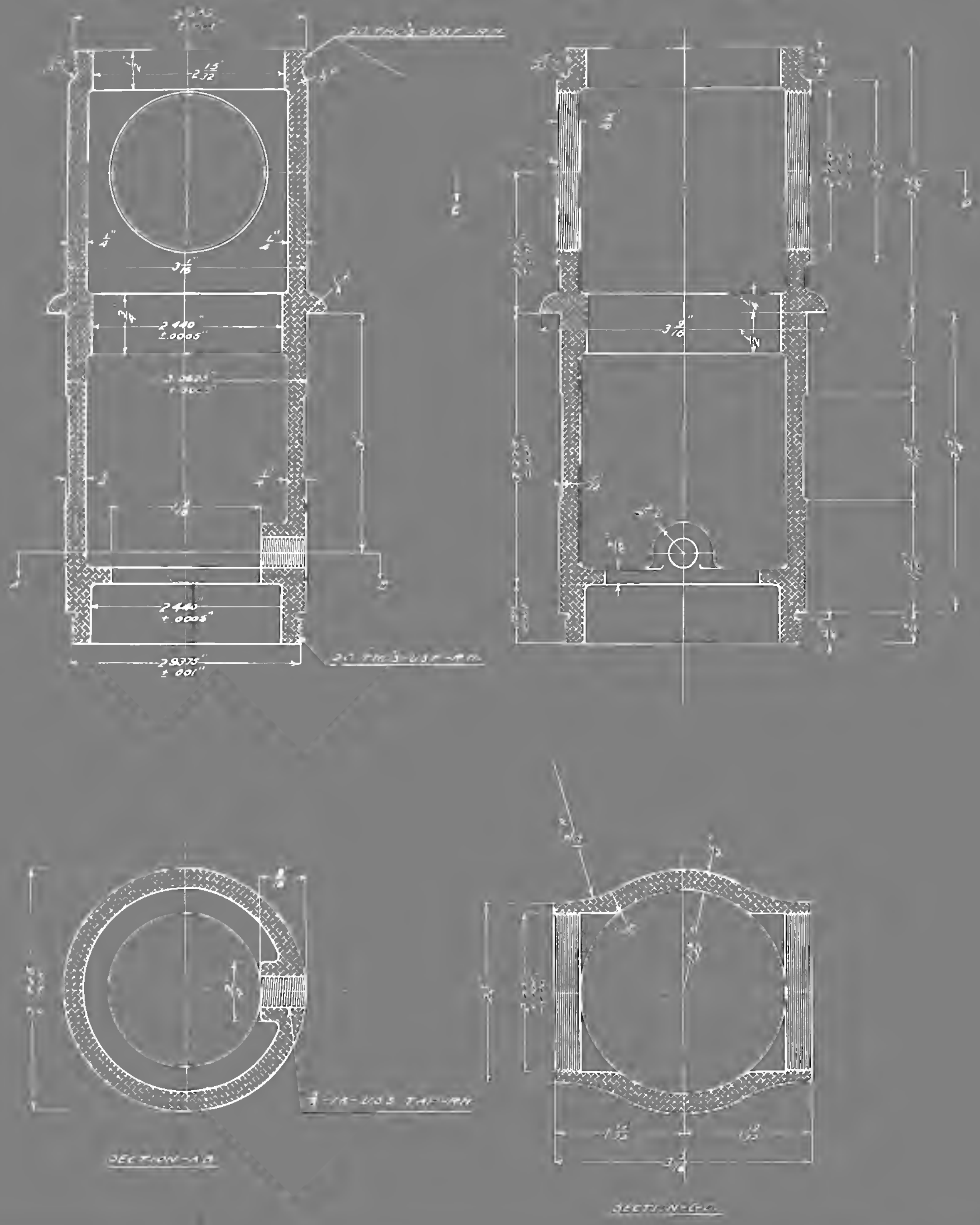
DESIGN OF
200 H.P. AERONAUTICAL ENGINE

SCALE FULL SIZE

DESIGNERS:
P.E. WINKLESS
GEO. PETERSON
C.F. PETERSON

APPROVED BY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

THESIS - 1920
DRWG. NO. 16


2 15/32
1/4"
1/4"
3 1/16"
2.440"
±.0005"
3.0625"
2.440"
±.0005"
2.9375"
±.001"
20 THD'S-USF-RH
20 THD'S-USF-RH
3 1/8"
3/8-16-USS TAP-RH
1 9/16
1 9/16
3 1/4
SECTION-A-B
SECTION-C-D

VERTICAL DRIVE HOUSING.

1-REQ'D. — MATERIAL: ALUMINUM ALLOY.
ANALYSIS
ALUMINUM - 97%
MAGNESIUM - 3%

DESIGN OF
200 H.P. AERONAUTICAL ENGINE

SCALE: FULL SIZE

DESIGNERS:
F.E. WINKLESS
G.E.O. PETERSON
C.F. PETERSON

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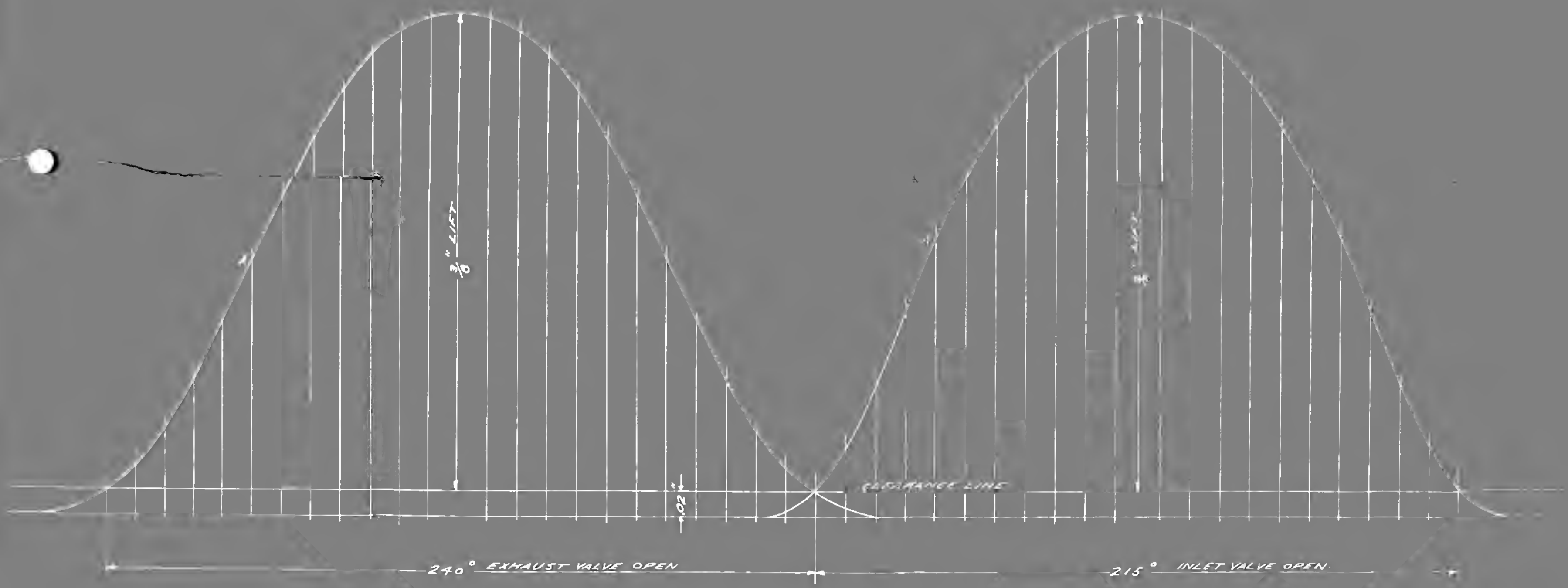
CHICAGO, ILL.

THESIS - 320
DWG. No. 16



3/8" LIFT
.02"
CLEARANCE LINE
240° EXHAUST VALVE OPEN
215° INLET VALVE OPEN

VALVE LIFT DIAGRAMS

SCALES:

ABSCISSAE: 1/8" = 1° CRANKSHAFT TRAVEL

ORDINATES: 16 [illegible] LIFT

DESIGN OF

200 H.P. AERONAUTICAL ENGINE

DESIGNERS

F.E. WINKLESS

E.E. PETERSON

[illegible]

APPROVED BY

ARMOUR INSTITUTE OF TECHNOLOGY

CHICAGO, ILL.

THESIS [illegible] JAN. [illegible]